AN INVESTIGATION OF THE DESIGN PARAMETERS OF PRESSURE FIRED MARINE BOILERS

BY

Lieutenant John R. White, USN

Submitted in Partial Fulfillment of the Requirements for the Master of Science Degree in Naval Architecture and Marine Engineering and the Professional Degree, Naval Engineer, at the Massachusetts Institute of Technology.

17 May, 1963

Thesis Supervisor: Commander J. R. Baylis, USN,
Associate Professor of Marine Engineering.

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ABSTRACT

The minimum total weight of propulsion machinery plus fuel for a ship of given cruising speed, top speed, and endurance must be obtained by finding the optimum balance of machinery weight and efficiency. The pressure fired, or supercharged boiler is considered herein as a substitute for the more conventional forced draft boiler. This paper is not concerned with the design of a particular boiler. Its purpose is to find the optimum values of furnace pressure, gas pressure drop through the boiler, and steam pressure for pressure fired boilers when such boilers are integrated with modern destroyer steam plants. The optimum furnace pressure is found to be about 30 psia at the cruising steam rate, which corresponds to 100 psia at the full power steam rate. Gas pressure drops at optimum conditions are about 4.8 psi and 20 psi at cruising and full power respectively. The best steam pressure is found to be slightly higher than the optimum for conventional boilers. The steam flow rate and the condenser pressure at full load also exhibit higher optimums. The optimum supercharged boiler weighs about 60% of the corresponding forced draft boiler. The entire plant plus fuel for the cycle incorporating supercharged boilers weighs about 90% of the conventional plant plus fuel, representing a saving of from 100 to 160 tons, depending on the cruising range.

Consideration is also given to the tube metal temperatures, steam velocities and pressure drops, and circulation in an optimum boiler, showing that these do not place serious limitations on designs in this region.

Thesis Supervisor: Commander J. R. Baylis, USN.

Title: Associate Professor of Marine Engineering.

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NOMENCLATURE

A	projected surface area
A_c	cross sectional area
C_p	specific heat at constant pressure
D, d	diameter
G_b	steam flow through boiler in lb/hr
h	enthalpy
Δh	difference in, or gain or loss of, enthalpy
K	ratio of specific heats
k	constant
p	pressure
p_s	steam pressure
Δp	gas pressure drop through boiler, or $p_6 - p_2$
S	surface area
T	absolute temperature
U	heat conductance in BTU/hr ft^2 °R
V	velocity
v	specific volume
W	weight
w	flow rate in lb/hr
ρ	density

ACKNOWLEDGEMENT

The author expresses appreciation for the guidance and suggestions provided by Commander J. R. Baylis, USN, throughout the preparation of this thesis.

INTRODUCTION

A perennial problem facing the designer of marine propulsion plants is that of providing adequate power to meet speed and endurance requirements, while taking up a minimum of space and weight. The importance of reducing the weight of propulsion systems is pointed out very strikingly by Mandel (1). The speed advantages that destroyers and other high speed ships hold over slower types are entirely due to increases in the SHP/ton ratio, rather than improvements in the hull lift-to-drag ratios. In fact, the lift-to-drag ratios are actually less for the high speed ships. In comparing two extremes, a tanker and a hydrofoil craft, Mandel shows that the lift-drag ratio of the hydrofoil is two orders of magnitude less than that of the tanker, but its SHP/ton ratio is three orders of magnitude larger. The predominant factor contributing to increased SHP/ton ratios is low specific weight power plants. Furthermore, wave making drag does not present an insurmountable barrier to increased speeds. The principal barrier to increased ship speeds is power plant weight. It is estimated that if a conventional 2000 ton destroyer hull were provided with the light weight power plant envisioned for hydrofoil boats, its top speed would be about 65 knots (1).

Of more universal interest to shipowners and operators is the increased payload capacity realized by utilizing light weight, compact power plants. The amount of fuel needed to meet endurance requirements must be charged against the weight and volume of the propulsion system. Efficiency of the plant therefore assumes an importance of equal magnitude to the weight of the hardware itself. Unfortunately, low specific weight and high efficiency normally run inversely to one another, so an optimum balance must be found to yield the least weight and/or volume of propulsion machinery plus fuel for a given set of performance criteria.

The optimization of several design parameters of a modern destroyer

propulsion plant was undertaken by Hayman and Otto (2). The DLG-6 power plant was used as a model, and the parameters optimized as a function of range were boiler steam pressure, condenser pressure, condenser surface area, low pressure turbine exhaust annulus area, and low pressure turbine leaving loss. This paper may be thought of as an extension of Hayman and Otto's work by the addition of one more cycle parameter, boiler furnace pressure. This is done by substituting a pressure fired, or supercharged boiler in the cycle in place of the conventional forced draft boiler. The terms "pressure-fired" and "supercharged" appear to be used interchangeable in the literature, as are the words "boiler" and "steam generator". Some manufacturers prefer to use "supercharged steam generator", others refer to a "pressure-fired boiler", and so forth. In this paper, any combination of these names means only one thing, a device designed to produce steam, the exhaust gases of which drive a supercharging unit, which in turn pressurizes the firebox.

Probably the earliest application of pressure fired boilers was the Velox cycle, developed by Brown-Boveri in the early '30's. A more recent application is a boiler designed by the Foster Wheeler Corporation, and recently tested and evaluated by the United States Navy on a test stand ashore. The Foster Wheeler boiler uses natural circulation, while the Velox design is for forced circulation. The advantages and disadvantages of supercharged boilers are well outlined by Frankel (3), Mills (4), and others, and will not be repeated here.

When the gas turbine is designed to produce a net output of work, the cycle is best thought of as a combined steam-gas cycle. However, in this analysis it is assumed that the gas turbine serves no other purpose then to drive the compressor. The supercharger unit may then be considered to be merely an accessory to the boiler, and the supercharged boiler may be substituted as a package for the conventional boiler and its associated forced draft blowers in the destroyer cycle analyzed by Hayman and Otto.

The Foster Wheeler design is used as a basic model for the boilers discussed in this paper. This boiler is well described by Daman (5), and Li Causi (6) gives some more information and a good cut-away picture. This design was used as a basis from which to scale constants for the weight equations, and as a check to insure that temperatures, flow rates, surface areas, and so forth, calculated in the analysis were reasonable. This study was kept independent of any particular boiler geometry. Its purpose was to find the optimum values of certain design parameters for any pressure fired boiler, when such boiler is integrated with a modern destroyer plant. The results indicate the direction that future experimentation in this field should take.

PROCEDURE

The basic assumptions made by Hayman and Otto will be retained in order to enable a direct comparison of the pressure-fired boiler to the conventional boiler. These assumptions are:

(1) DLG-6 plant used as basic model

(2) Top steam temperature 1050°F

(3) SHP full power = 85000

(4) SHP cruising = 12, 100.

The cycle parameters varied in (2) were:

(1) boiler steam pressure

(2) condenser pressure

(3) low pressure turbine exhaust annulus area

(4) low pressure turbine leaving loss

(5) condenser surface.

The part of Hayman's and Otto's study used in this paper is the tabulation of machinery weights (less boiler) for each steam pressure and condenser pressure. This consists of table IV in (2).

The variables used in the pressure fired boiler analysis are:

(1) furnace gas pressure (p_4)

(2) gas flow rate (w_g)

(3) gas pressure drop through boiler (Δp)

(4) temperature of gas at exit from generating tube bank and entrance to gas turbine (T_6)

(5) steam pressure (p_s)

The boiler is optimized at cruising rate, or 20 knots. The total steam flow is assumed to be a constant 14.4% of full power steam flow. Cruising is

normally done on two boilers instead of the four required for full power, so cruising steam flow per boiler is 28.8% of full power flow per boiler.

A preliminary assumption was made that the best value of T_6 would be that temperature which would result in a self sustaining supercharger with no excess gas by-passed. A temperature lower than the ideal T_6 would require outside power to be applied to the supercharger compressor in order to maintain the required air flow and pressure to the firebox. A T_6 higher than necessary for self-sustenance would mean that a portion of hot exhaust gases must be by-passed around the supercharger and either passed through an additional heat exchanger or dumped to the atmosphere. This would obviously result in increased plant weight and/or an unnecessary energy loss. The ideal T_6 would be a function of furnace pressure, gas side pressure drop, and supercharger efficiencies. These values are tabulated in table I. It was found, however, that in some cases the tabulated values of T_6 were too close to the temperature of the saturated steam in the generating tubes, and that the generating tube banks would have to be unduly large to cool the gases to T_6. In some cases the additional weight needed in the generating tubes and economizer was greater than the fuel weight saved by making the supercharger just self sustaining. The tabulated values of T_6 were then raised until the minimum weight of generating tubes plus economizer plus fuel was obtained.

A first law heat balance was then written for each value of p_4, Δp, and p_s, and for the two extreme boiler flow rates anticipated. The main purpose of the heat balance was to find the required gas flow per pound of steam evaporated (w_g/G_b). This is a measure of boiler efficiency and was used to obtain the weight of fuel required per mile.

Weight equations were developed as functions of the variable parameters. The constants for these equations were obtained from the Foster Wheeler boiler described by Daman (5) and Li Causi (6) and from existing conventional boilers.

Using the weight equations and the fuel weights from the heat balances,

TABLE I

GAS TURBINE ENTRANCE TEMPERATURES (T_6) AS FUNCTIONS OF P_4, ΔP, AND P_s.

P_4	ΔP	P_4	ΔP	A	B	C	D	E
				T_6				
CRUISING		FULL POWER		$P_s \rightarrow$	800	1000	1200	1400
19.4	0.30	40	2	1010	1020	1050	1115	1165
	0.60		4	1070	1070	1070	1070	1165
	1.20		8	1240	1240	1240	1240	1240
	1.80		12	1530	1530	1530	1530	1530
23.1	0.57	60	3	970	1020	1060	1115	1165
	1.13		6	1045	1045	1060	1115	1165
	2.26		12	1190	1190	1190	1190	1190
	3.40		18	1395	1395	1395	1395	1395
26.8	0.86	80	4	895	1020	1060	1115	1165
	1.73		8	1061	1061	1061	1115	1165
	3.46		16	1190	1190	1190	1190	1190
	5.20		24	1405	1405	1405	1405	1405
30.5	1.19	100	5	977	1020	1060	1115	1165
	2.40		10	1040	1040	1060	1115	1165
	4.80		20	1105	1105	1105	1115	1165
	7.20		30	1465	1465	1465	1465	1465
34.3	1.52	120	6	1020	1020	1060	1115	1165
	3.05		12	1060	1060	1060	1115	1165
	6.10		24	1220	1220	1220	1220	1220
	9.10		36	1320	1320	1320	1320	1320
	12.20		48	1905	1905	1905	1905	1905
37.9	1.89	140	7	1020	1020	1060	1115	1165
	3.79		14	1085	1085	1085	1115	1165
	7.57		28	1220	1220	1220	1220	1220
	11.34		42	1485	1485	1485	1485	1485
	15.10		56	1985	1985	1985	1985	1985

COLUMN A LISTS T_6 REQUIRED FOR SELF-SUSTENANCE, COMPUTED FROM EQUATION (14 k). COLUMNS B THROUGH E LIST REVISED T_6 AS DESCRIBED IN THE PROCEDURE.

a certain value of Δp could be picked for each p_4, p_s, and range that would yield a minimum weight. These minimum weights were plotted versus p_4 for each p_s, range, and for the two extreme boiler flows as figures I through VIII. Each of these curves yield a minimum weight of boiler plus fuel for a particular p_4. The data from (2) can be introduced at this point. The table of machinery variable weights as functions p_s and G_b can be combined with the minimum boiler plus fuel weights of figures I through VIII. Linear interpolation is used for values of G_b between the two extremes. Interpolation curves of boiler weight versus G_b for each p_s and R are shown in figures IX through XII. The result of this are figures XIII through XVI, plots of total weight versus boiler flow rate for each p_s and range. A certain G_b for each p_s and range yields a minimum weight. These minimums are plotted in figures XVII through XX, weight versus steam pressure for each range. Finally, figure XXI may be drawn, which shows the curve of minimum weight as a function of range and steam pressure.

RESULTS

The principal results of this study are the values of the design parameters for a pressure fired boiler that yield the least weight solution for the summation of propulsion machinery and fuel, when such boiler is integrated in a modern destroyer steam plant. Secondary results are the optimum values of cycle parameters, such as steam flow rate and condenser pressure, which are affected by the substitution of a pressure fired boiler for the heavier forced draft boiler. The comparison of weights of the two plants, one with pressure fired boilers and the other with conventional boilers, is also of some interest.

All of the optimization calculations performed were done with respect to least total weight at cruising power. However, the values of furnace pressure (p_4) shown on curves I through VIII is the corresponding full power pressure. These curves show that the optimum furnace pressure is in all cases very close to 100 psia full power, or 30.5 psia cruising power pressure. The optimum gas pressure drop through the boiler is about 20% of the full power furnace pressure. The relationships between full power and cruising rates are as shown in the details of procedure.

Figures XIII through XVI show that as steam pressures increase, the minimum point in the weight curves move to lower steam rates. These lower steam rates correspond to lower condenser pressures. The effect of steam pressure on optimum condenser pressure is shown in figure XXII.

Figure XXI is the plot of optimum steam pressure as a function of range. The dotted curve is optimum steam pressure for the plant with conventional boilers, and was taken from figure XXVI of reference (5).

Total weights of propulsion machinery plus fuel are listed in table III.

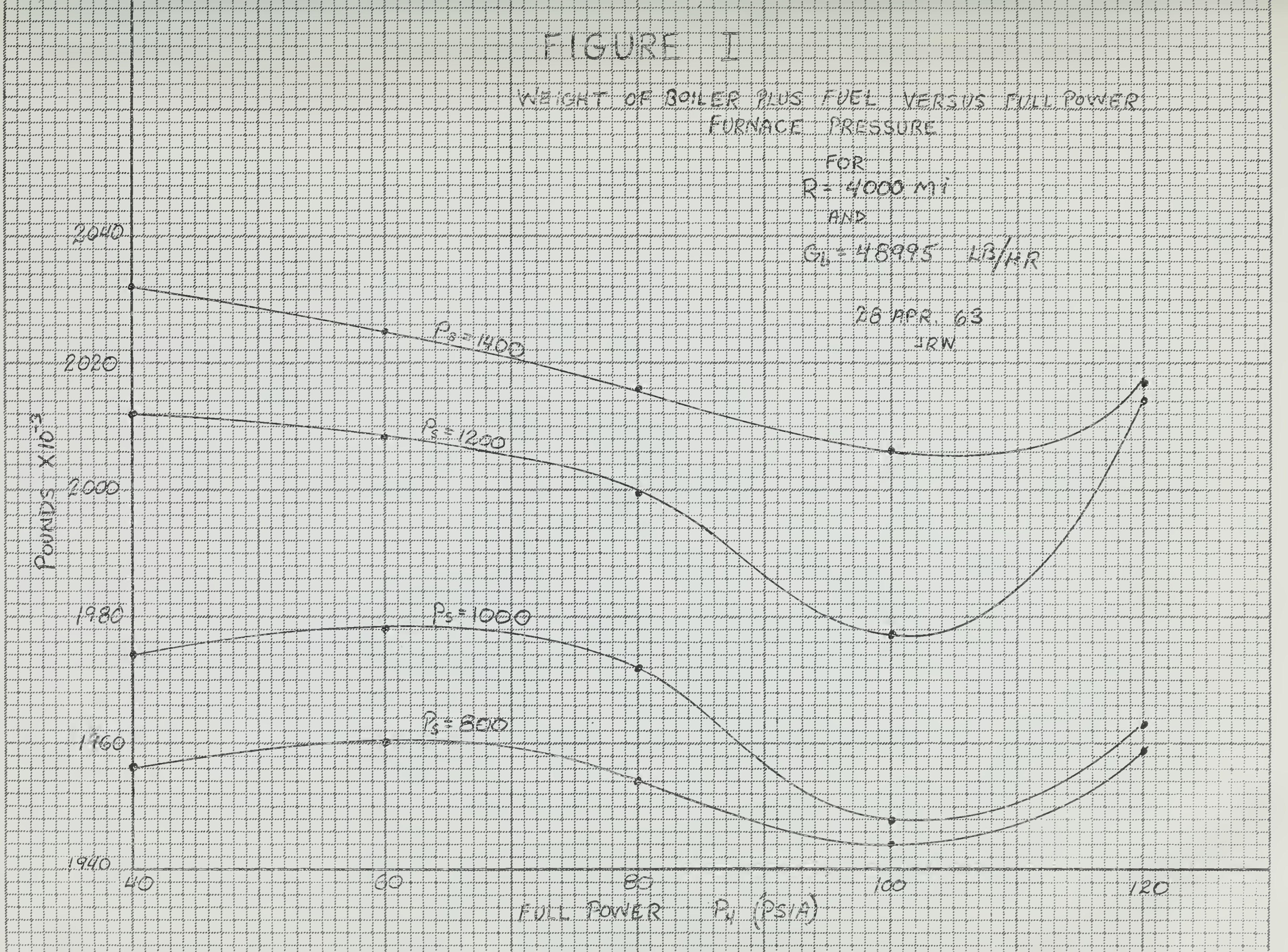
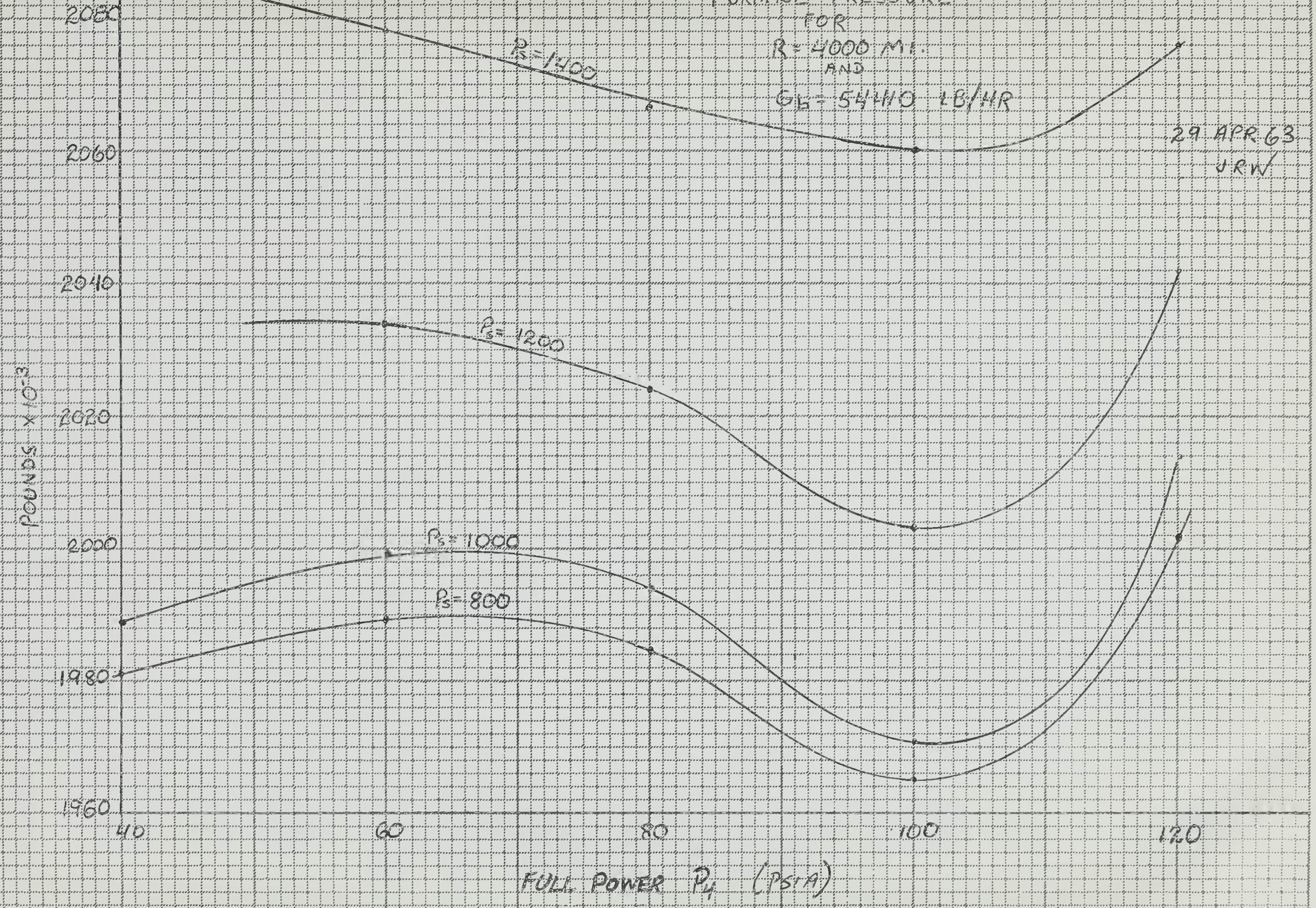


FIGURE I
WEIGHT OF BOILER PLUS FUEL VERSUS FULL POWER
FURNACE PRESSURE
FOR
R = 4000 mi
AND
G_L = 48995 LB/HR
28 APR. 63
JRW
P_s = 1400
P_s = 1200
P_s = 1000
P_s = 800
2040
2020
2000
1980
1960
1940
POUNDS X10^{-3}
40
60
80
100
120
FULL POWER P_1 (PSIA)

FIGURE II

WEIGHT OF BOILER PLUS FUEL VERSUS FULL POWER FURNACE PRESSURE FOR R = 4000 MI. AND G_b = 54410 LB/HR



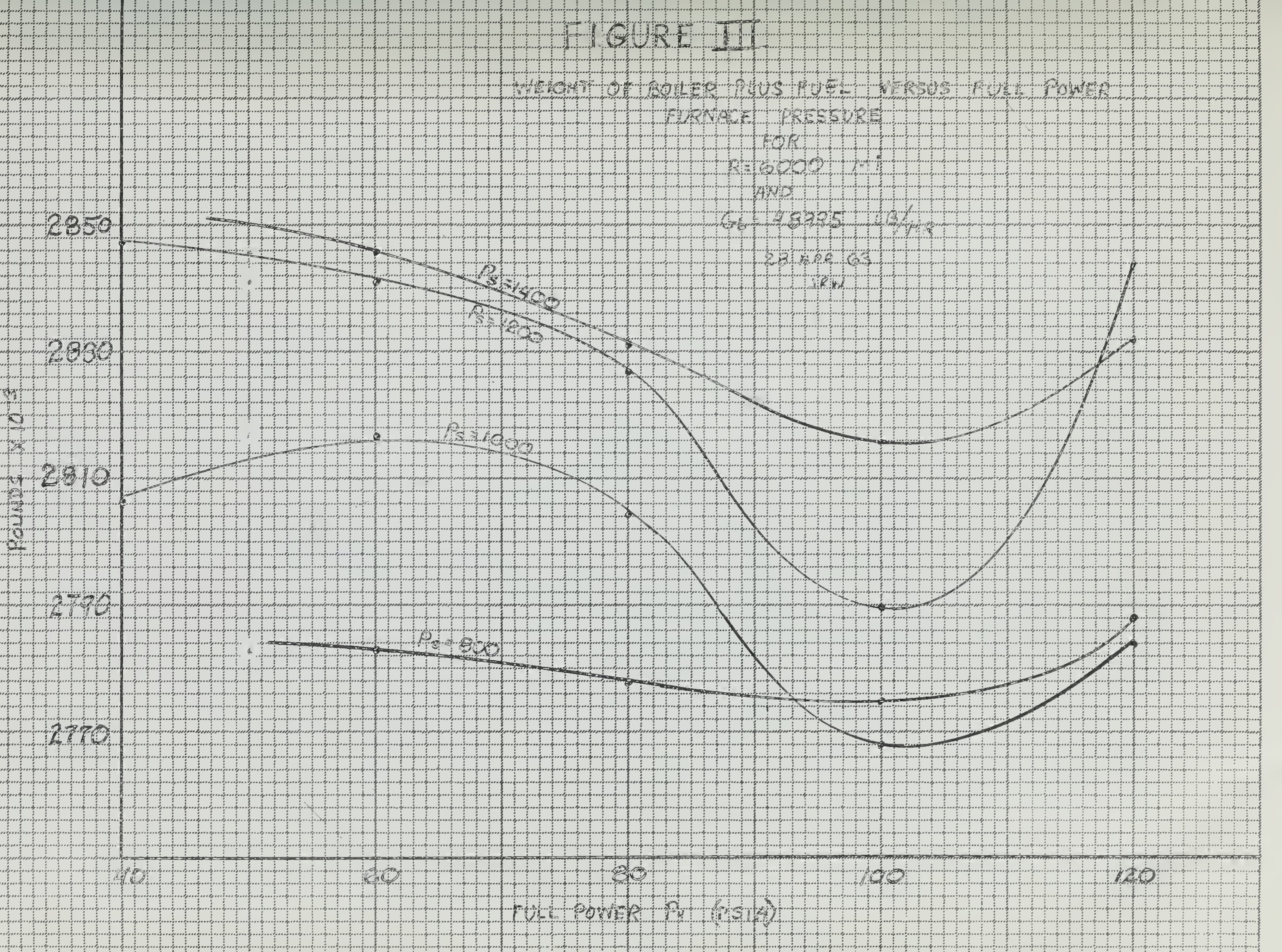
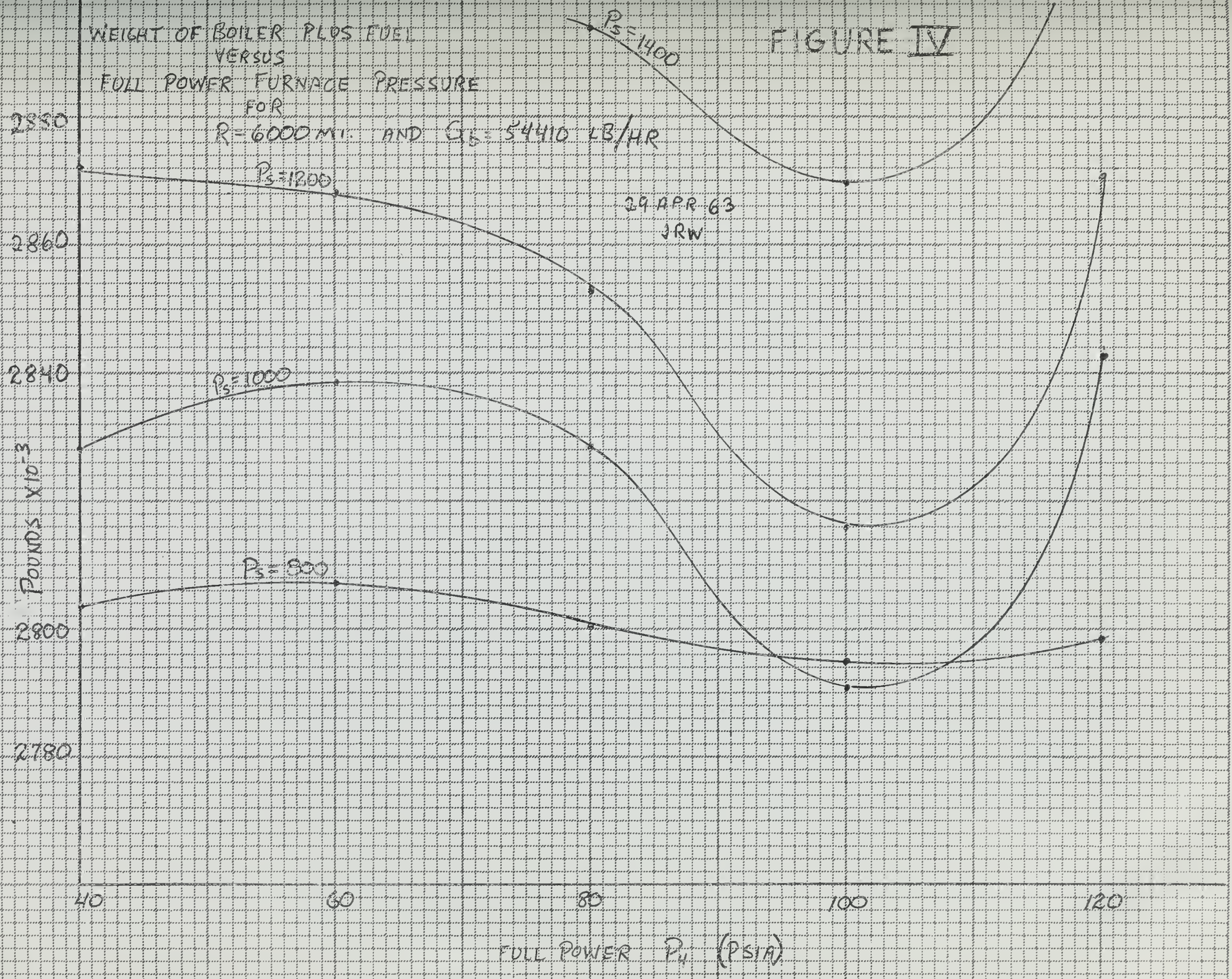


FIGURE III

WEIGHT OF BOILER PLUS FUEL VERSUS FULL POWER
FURNACE PRESSURE
FOR
R=6000 MI
AND


WEIGHT OF BOILER PLUS FUEL
VERSUS
FULL POWER FURNACE PRESSURE
FOR
R = 6000 MI. AND G_b = 54410 LB/HR
29 APR 63
JRW
P_s = 1400
P_s = 1200
P_s = 1000
P_s = 800
2880
2860
2840
2800
2780
POUNDS $\times 10^{-3}$
40
60
80
100
120
FULL POWER P_4 (PSIA)
FIGURE IV

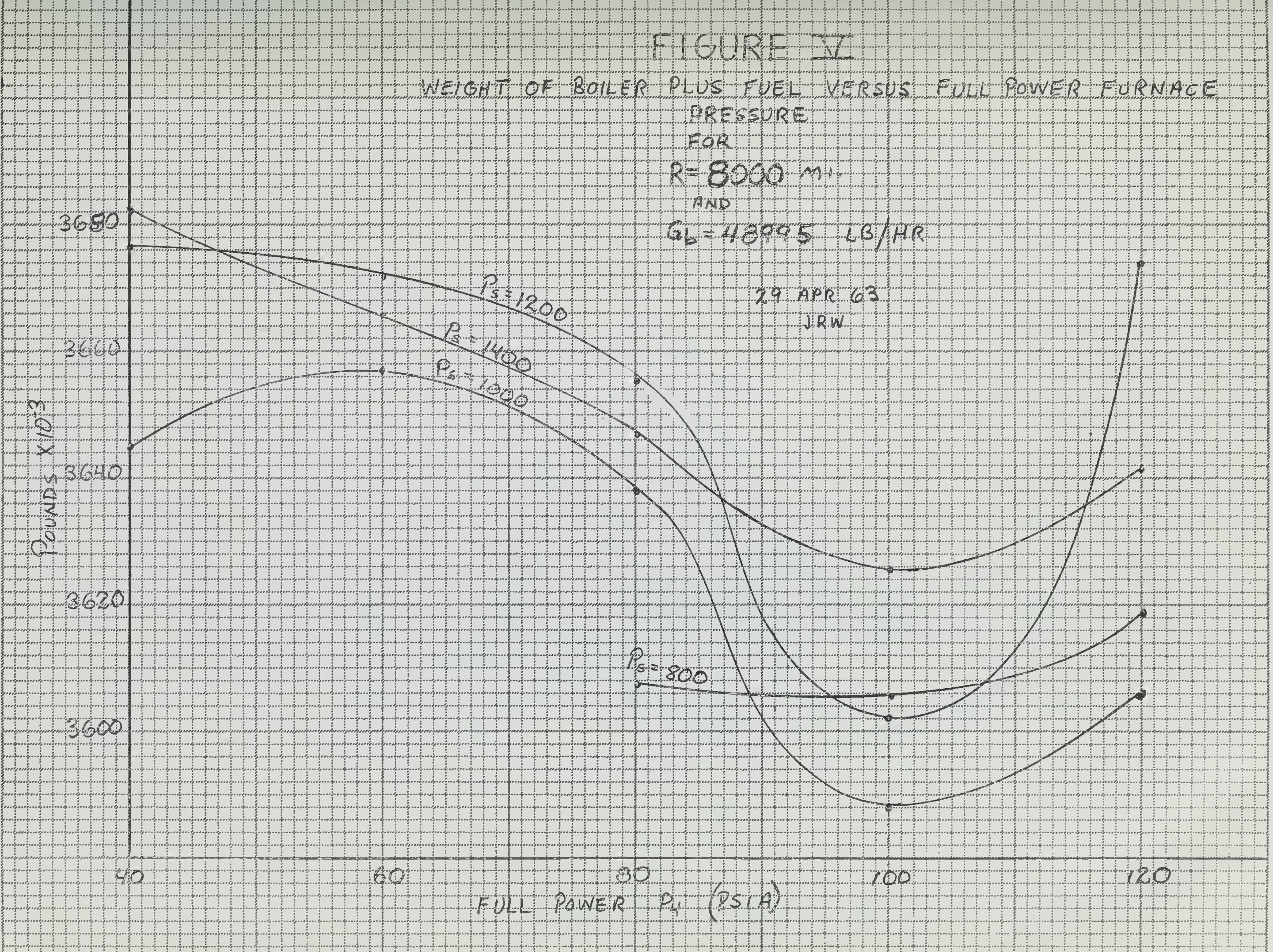


FIGURE V
WEIGHT OF BOILER PLUS FUEL VERSUS FULL POWER FURNACE PRESSURE
FOR
R= 8000 MI.
AND
G_b = 48995 LB/HR
29 APR 63
JRW
P_s = 1200
P_s = 1400
P_s = 1000
P_s = 800
3680
3660
3640
3620
3600
POUNDS X 10^{-3}
40
60
80
100
120
FULL POWER P_4 (PSIA)

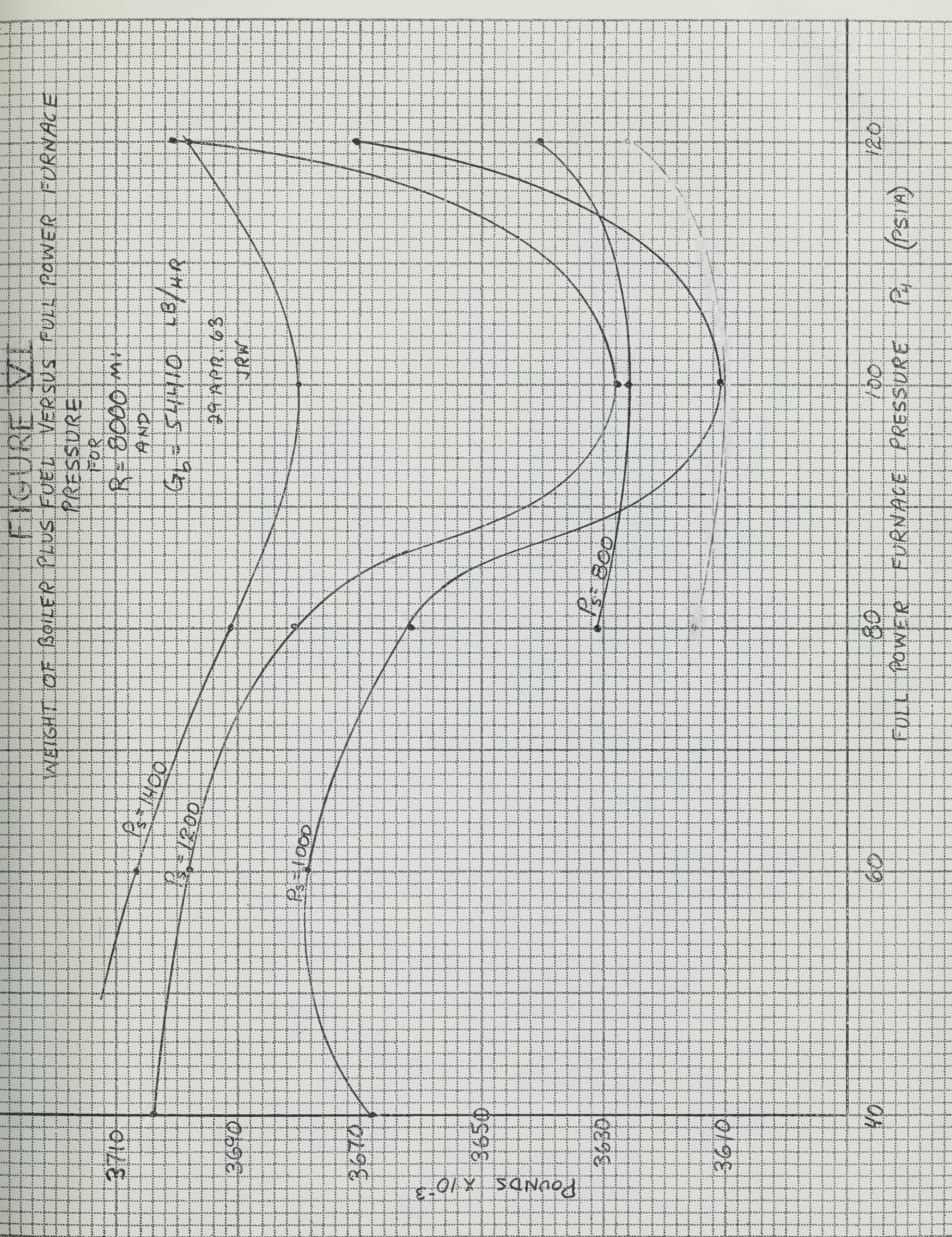


FIGURE VI
WEIGHT OF BOILER PLUS FUEL VERSUS FULL POWER FURNACE PRESSURE
FOR
R = 8000 MI
AND
G_b = 541410 LB/HR
29 APR. 63
JRW
P_s = 1400
P_s = 1200
P_s = 1000
P_s = 800
3710
3690
3670
3650
3630
3610
POUNDS X 10^{-3}
40
60
80
100
120
FULL POWER FURNACE PRESSURE P_4 (PSIA)

FIGURE VII

WEIGHT OF BOILER PLUS FUEL VERSUS FULL POWER FURNACE PRESSURE

FOR

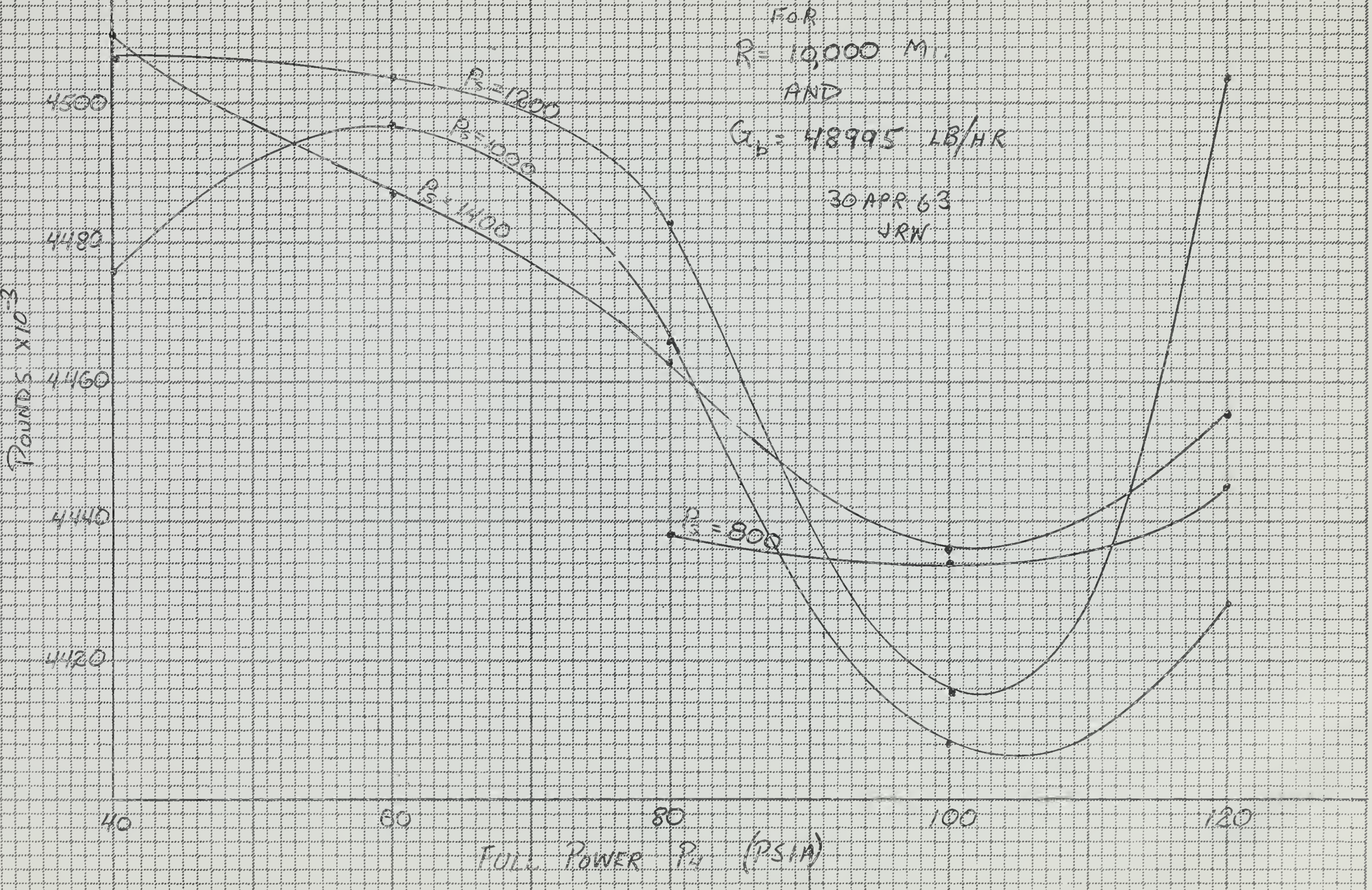
$R = 10{,}000$ MI.

AND

$G_b = 48995$ LB/HR

30 APR 63

JRW



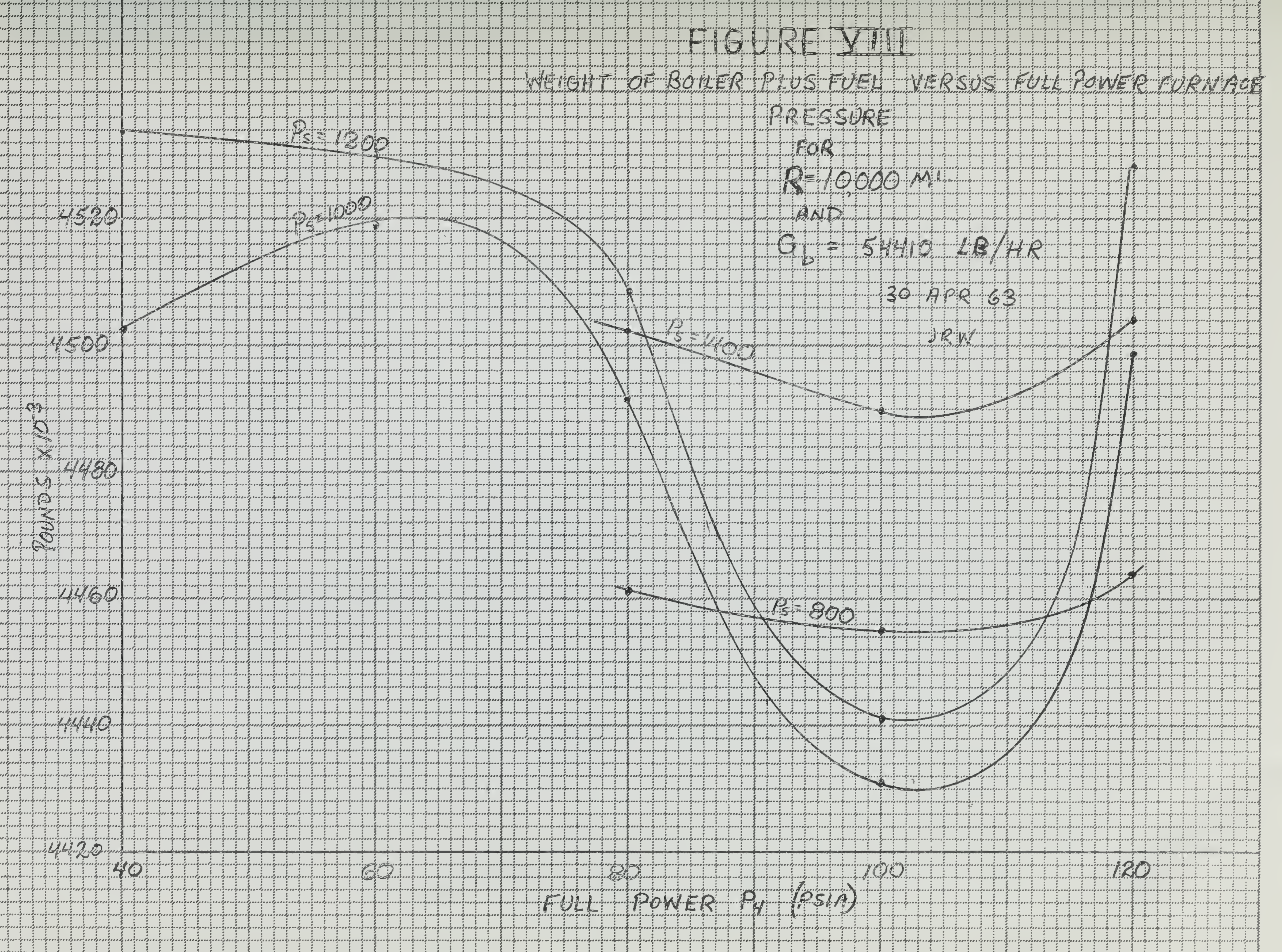


FIGURE VIII
WEIGHT OF BOILER PLUS FUEL VERSUS FULL POWER FURNACE PRESSURE
FOR
R = 10,000 MI.
AND
G_b = 54410 LB/HR
30 APR 63
JRW
P_s = 1200
P_s = 1000
P_s = 1400
P_s = 800
POUNDS X 10^3
4520
4500
4480
4460
4440
4420
40
60
80
100
120
FULL POWER P_4 (PSIA)

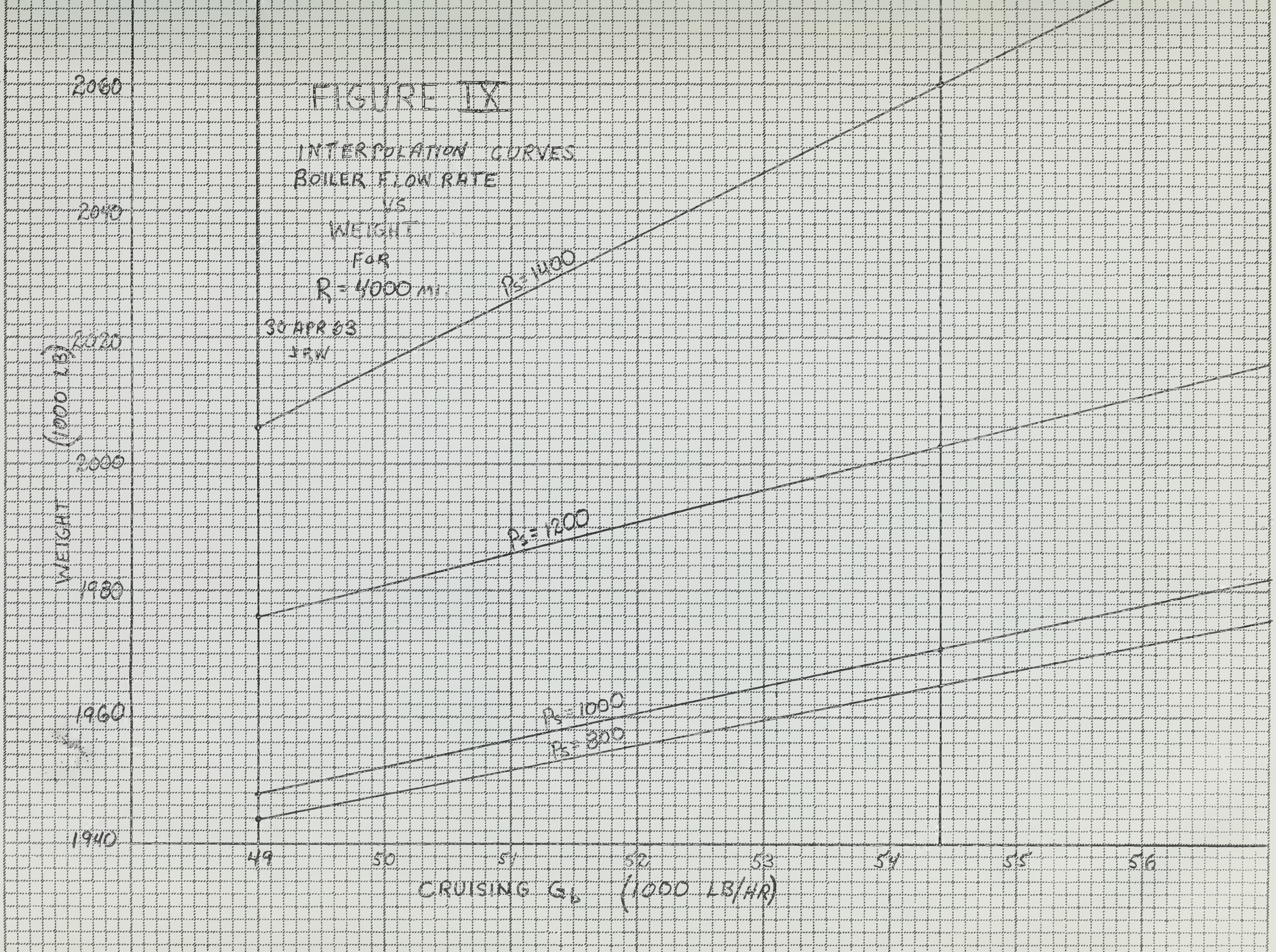


FIGURE IX
INTERPOLATION CURVES
BOILER FLOW RATE
VS
WEIGHT
FOR
R = 4000 mi.
30 APR 53
JRW
Ps = 1400
Ps = 1200
Ps = 1000
Ps = 800
WEIGHT (1000 LB)
2060
2040
2020
2000
1980
1960
1940
49
50
51
52
53
54
55
56
CRUISING Gb (1000 LB/HR)

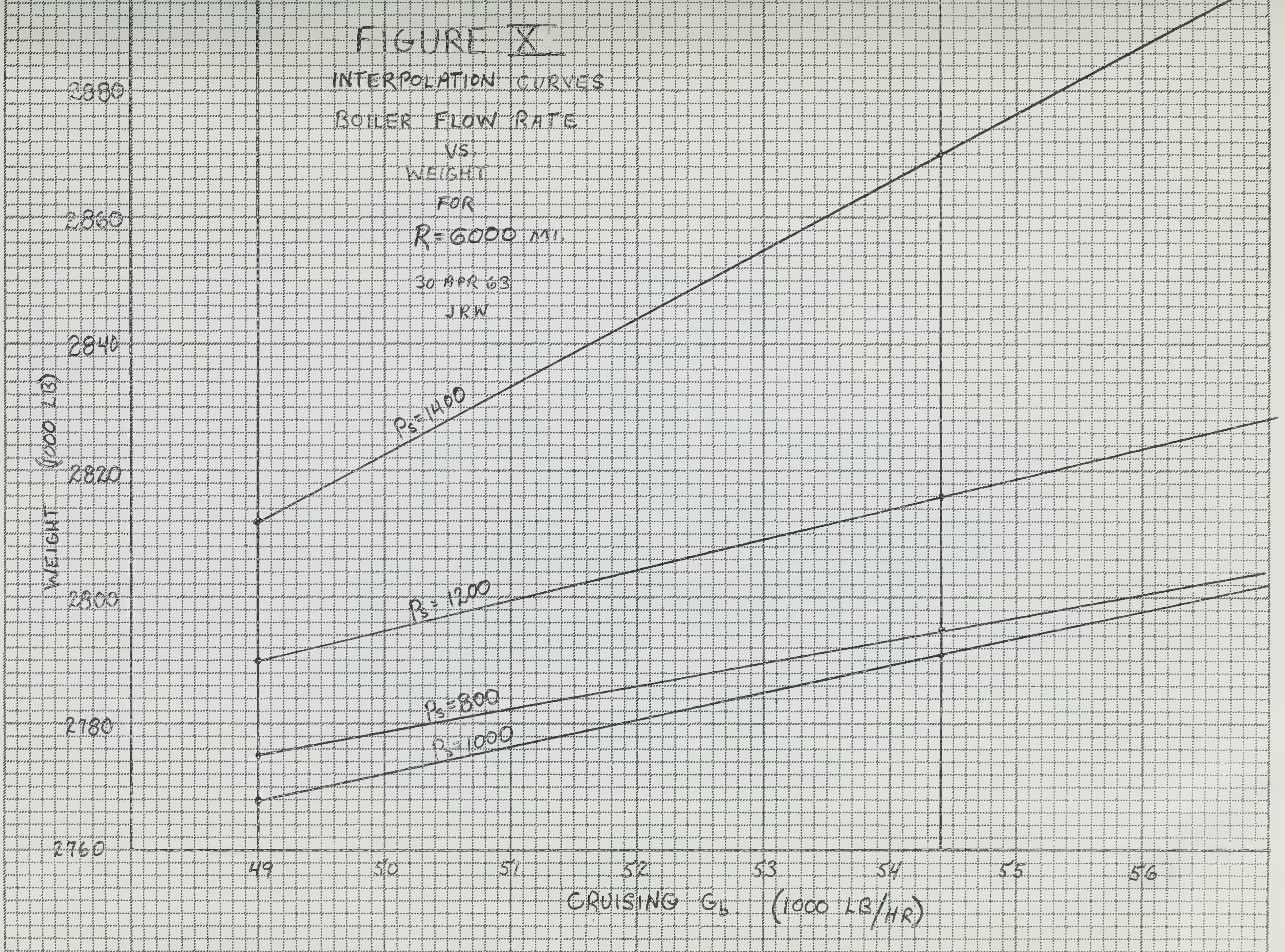


FIGURE X
INTERPOLATION CURVES
BOILER FLOW RATE
VS.
WEIGHT
FOR
R=6000 MI.
30 APR 63
JRW
2880
2860
2840
2820
2800
2780
2760
WEIGHT (1000 LB)
49
50
51
52
53
54
55
56
CRUISING Gb (1000 LB/HR)
Ps=1400
Ps=1200
Ps=800
Ps=1000

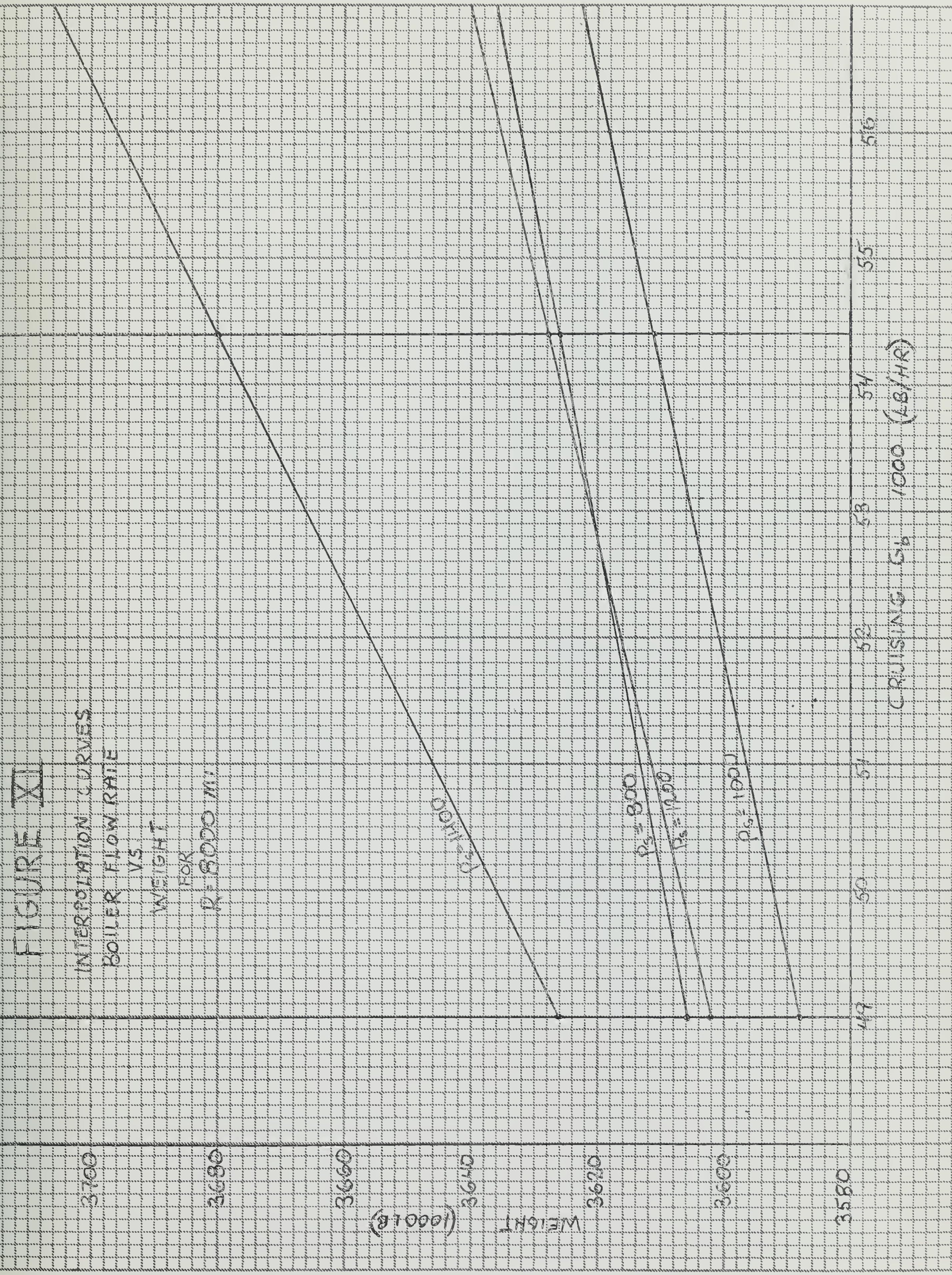


FIGURE XI
INTERPOLATION CURVES
BOILER FLOW RATE
VS.
WEIGHT
FOR
R = 8000 MI
$P_s = 1100$
$P_s = 800$
$P_s = 1200$
$P_s = 1000$
3580
3600
3620
3640
3660
3680
3700
WEIGHT (1000 LB)
49
50
51
52
53
54
55
56
CRUISING G_b 1000 (LB/HR)

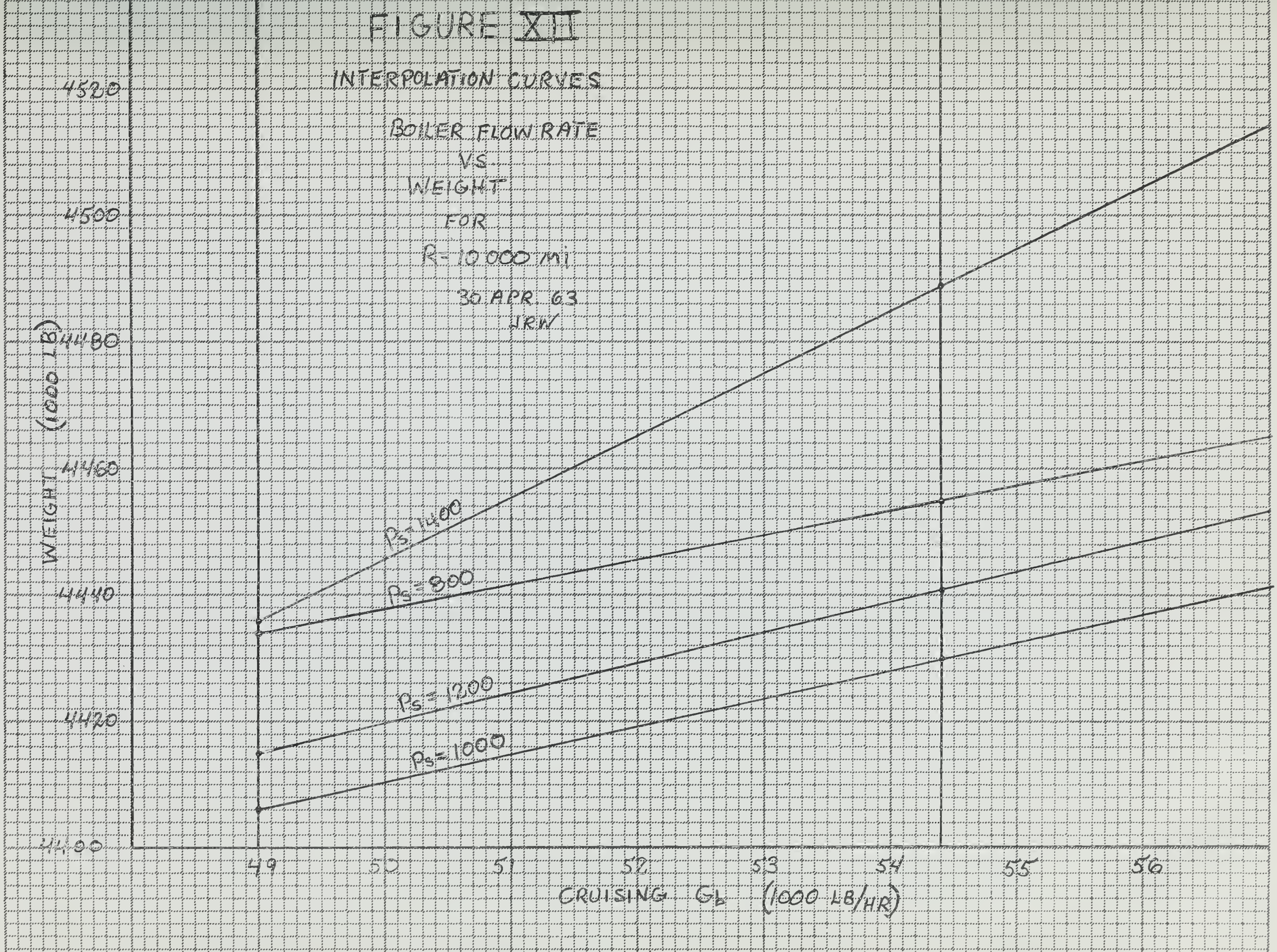


FIGURE XII
INTERPOLATION CURVES
BOILER FLOW RATE
VS
WEIGHT
FOR
R=10 000 mi
30 APR 63
JRW
4520
4500
4480
4460
4440
4420
4400
WEIGHT (1000 LB)
49
50
51
52
53
54
55
56
CRUISING G_b (1000 LB/HR)
P_s=1400
P_s=800
P_s=1200
P_s=1000

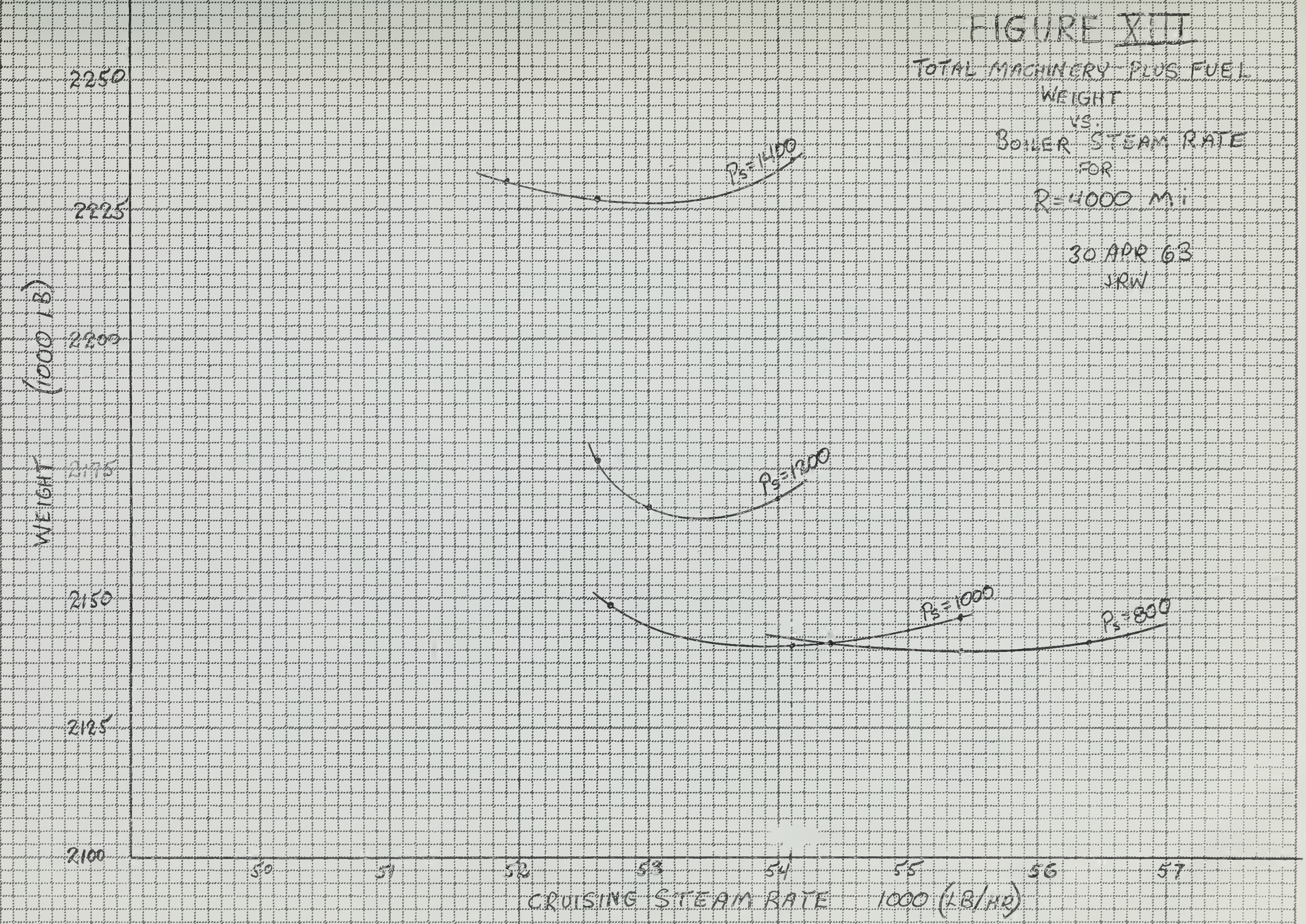


FIGURE XIII
TOTAL MACHINERY PLUS FUEL
WEIGHT
VS.
BOILER STEAM RATE
FOR
R=4000 Mi
30 APR 63
JRW
P_s=1400
P_s=1200
P_s=1000
P_s=800
2250
2225
2200
2175
2150
2125
2100
WEIGHT (1000 LB)
50
51
52
53
54
55
56
57
CRUISING STEAM RATE 1000 (LB/HR)

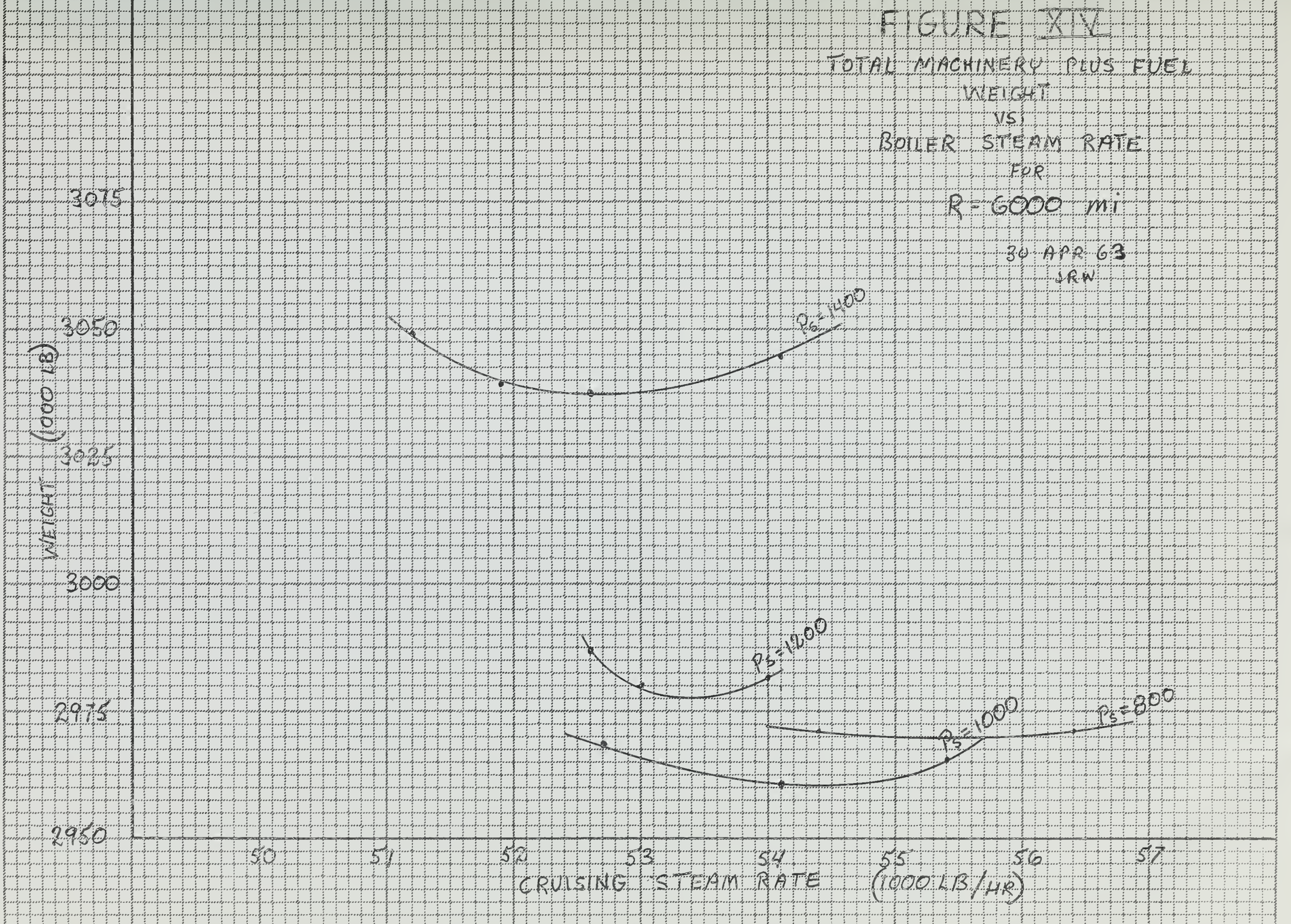


FIGURE XIV
TOTAL MACHINERY PLUS FUEL
WEIGHT
VS.
BOILER STEAM RATE
FOR
R = 6000 mi
30 APR 63
JRW
P_s=1400
P_s=1200
P_s=1000
P_s=800
3075
3050
3025
3000
2975
2950
WEIGHT (1000 LB)
50
51
52
53
54
55
56
57
CRUISING STEAM RATE (1000 LB/HR)

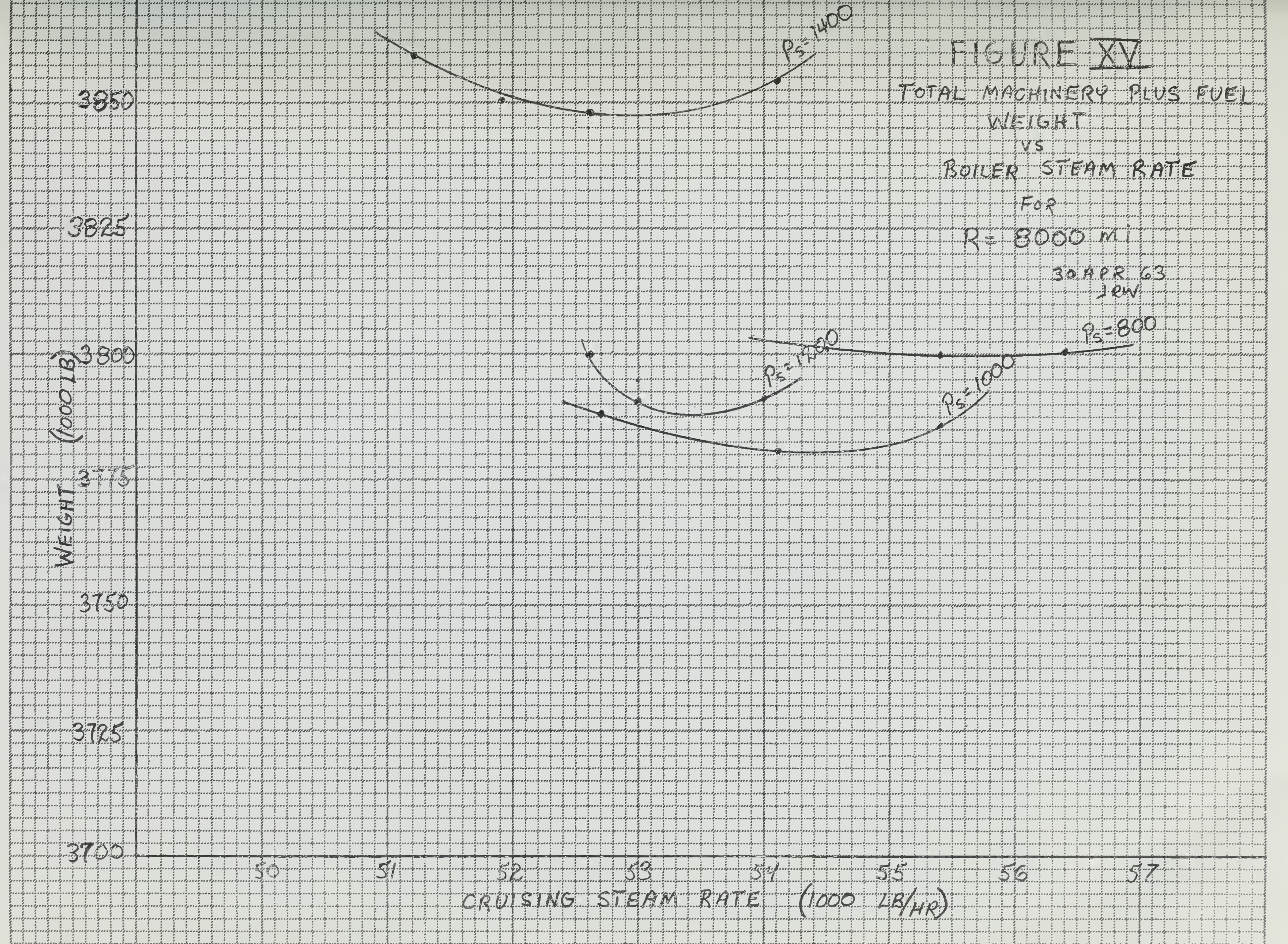


FIGURE XV
TOTAL MACHINERY PLUS FUEL
WEIGHT
VS
BOILER STEAM RATE
FOR
R = 8000 mi
30 APR 63
JRW
$P_s = 1400$
$P_s = 1200$
$P_s = 1000$
$P_s = 800$
3850
3825
3800
3775
3750
3725
3700
WEIGHT (1000 LB)
50
51
52
53
54
55
56
57
CRUISING STEAM RATE (1000 LB/HR)

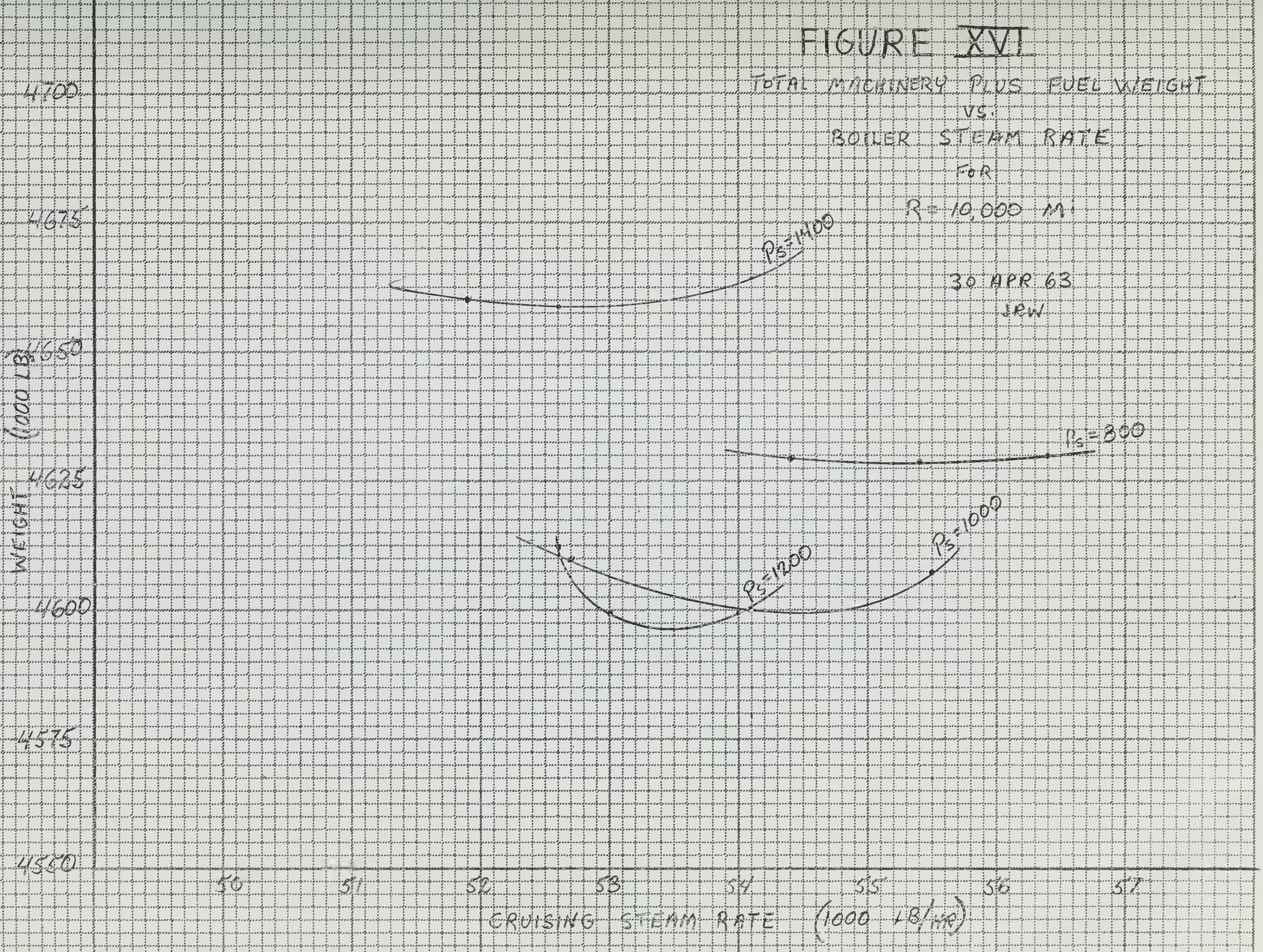


FIGURE XVI

TOTAL MACHINERY PLUS FUEL WEIGHT
VS.
BOILER STEAM RATE
FOR
R = 10,000 MI

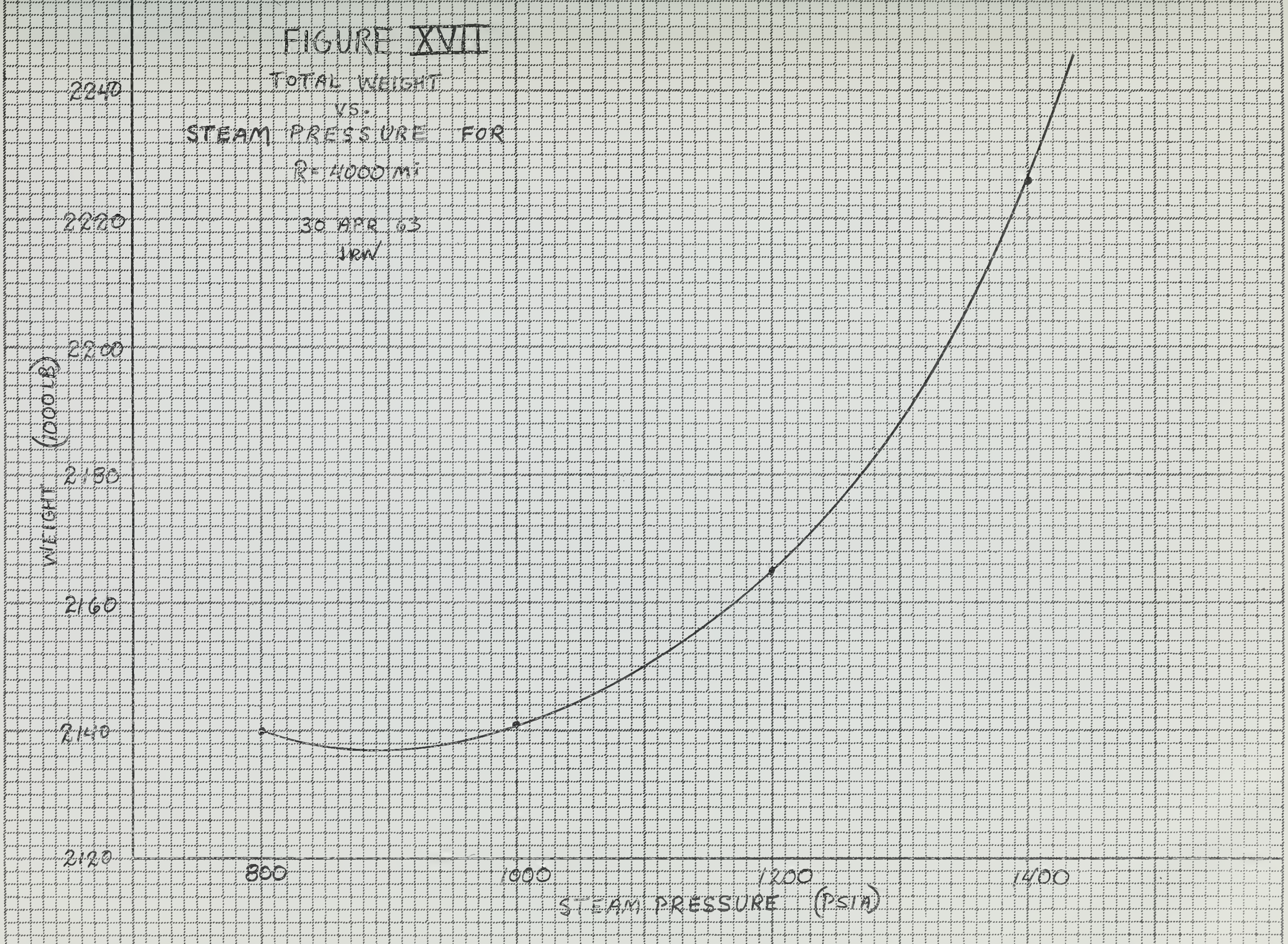


FIGURE XVII
TOTAL WEIGHT
VS.
STEAM PRESSURE FOR
R=4000 mi
30 APR 63
JRW
2240
2220
2200
2180
2160
2140
2120
WEIGHT (1000LB)
800
1000
1200
1400
STEAM PRESSURE (PSIA)

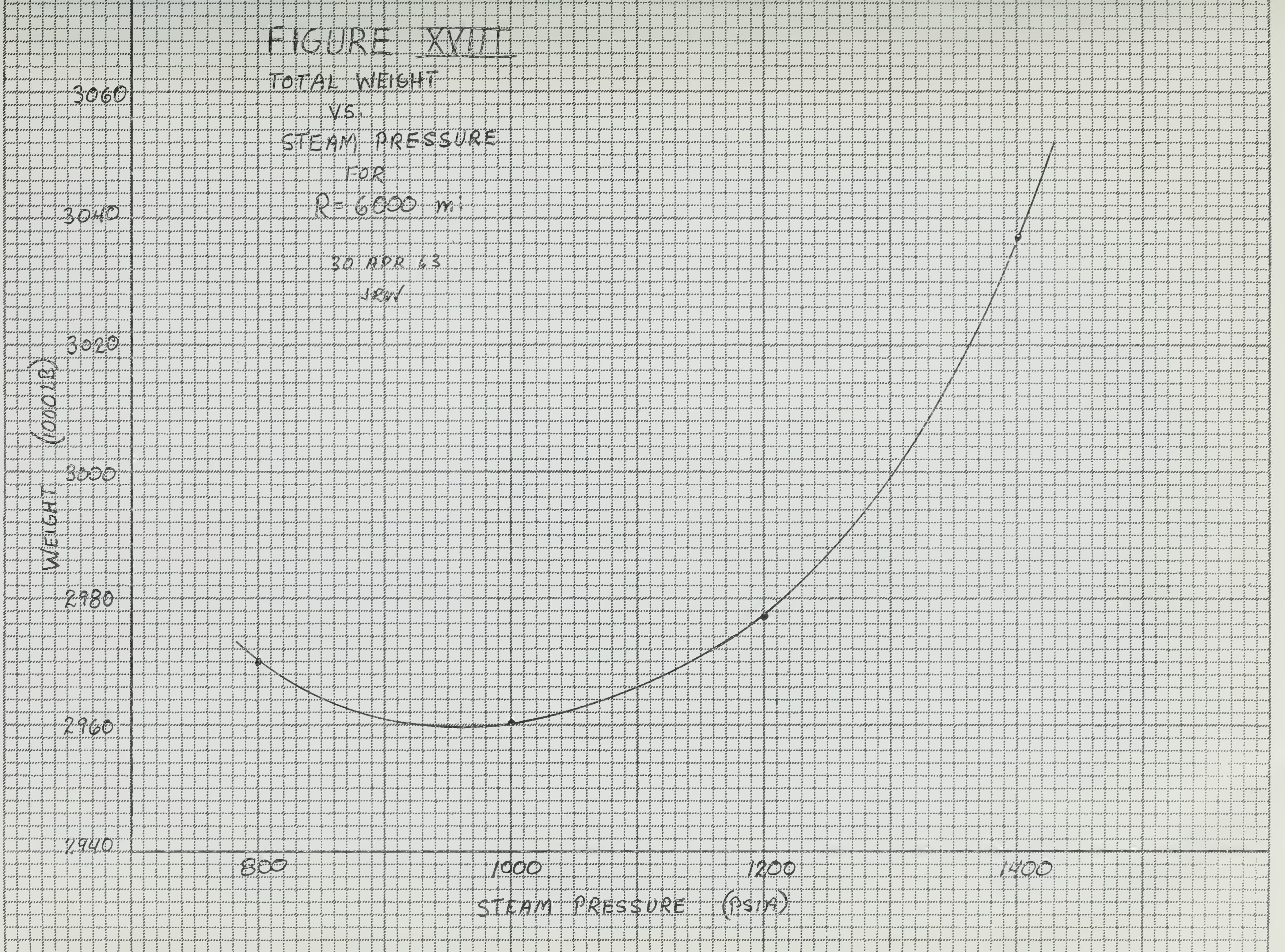


FIGURE XVIII
TOTAL WEIGHT
VS.
STEAM PRESSURE
FOR
R=6000 mi
30 APR 63
JRW
3060
3040
3020
3000
2980
2960
2940
WEIGHT (1000LB)
800
1000
1200
1400
STEAM PRESSURE (PSIA)

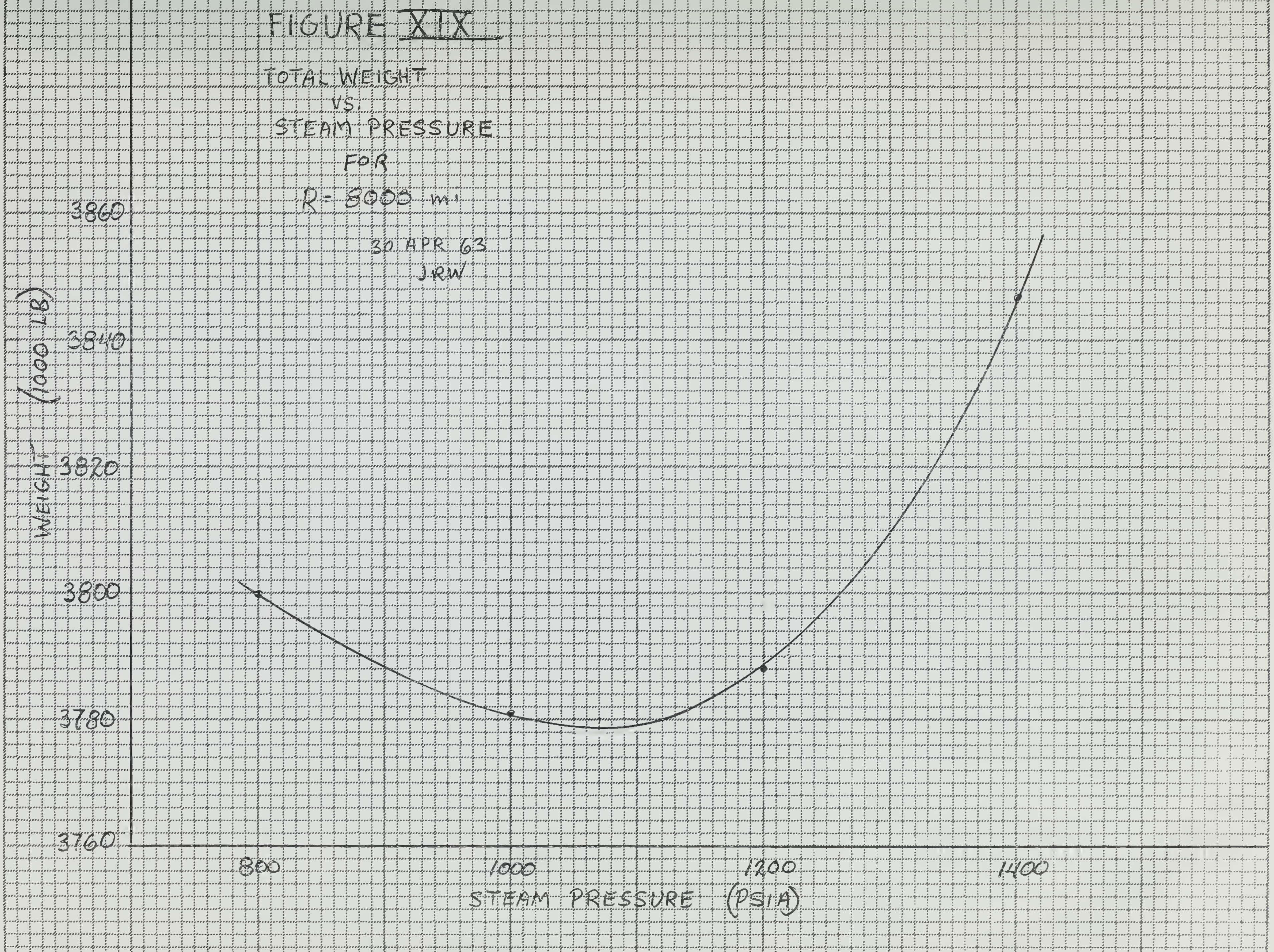


FIGURE XIX
TOTAL WEIGHT
VS.
STEAM PRESSURE
FOR
R = 8000 mi
30 APR 63
JRW
WEIGHT (1000 LB)
3860
3840
3820
3800
3780
3760
800
1000
1200
1400
STEAM PRESSURE (PSIA)

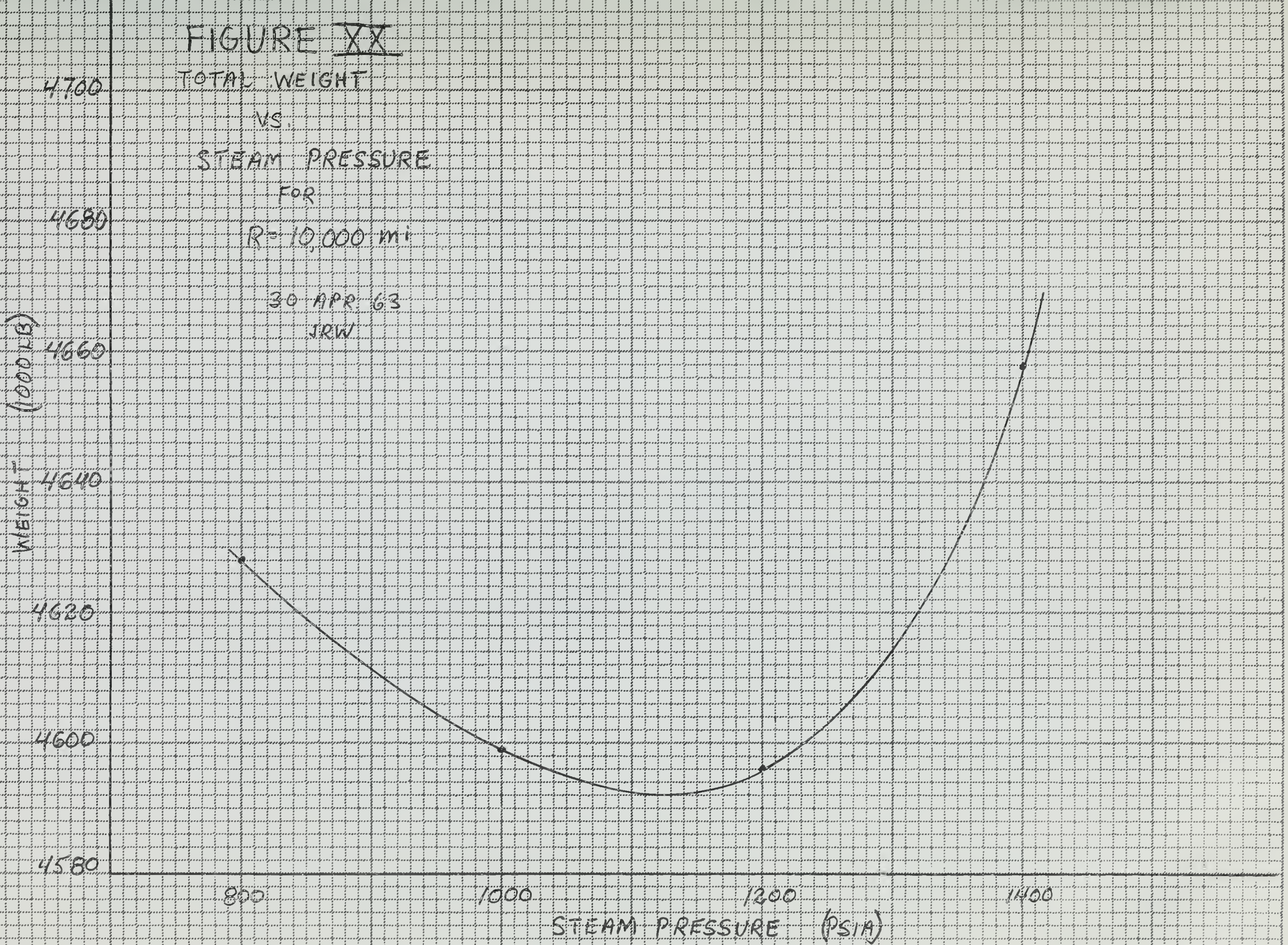


FIGURE XX
TOTAL WEIGHT
VS.
STEAM PRESSURE
FOR
R = 10,000 mi

30 APR 63
JRW

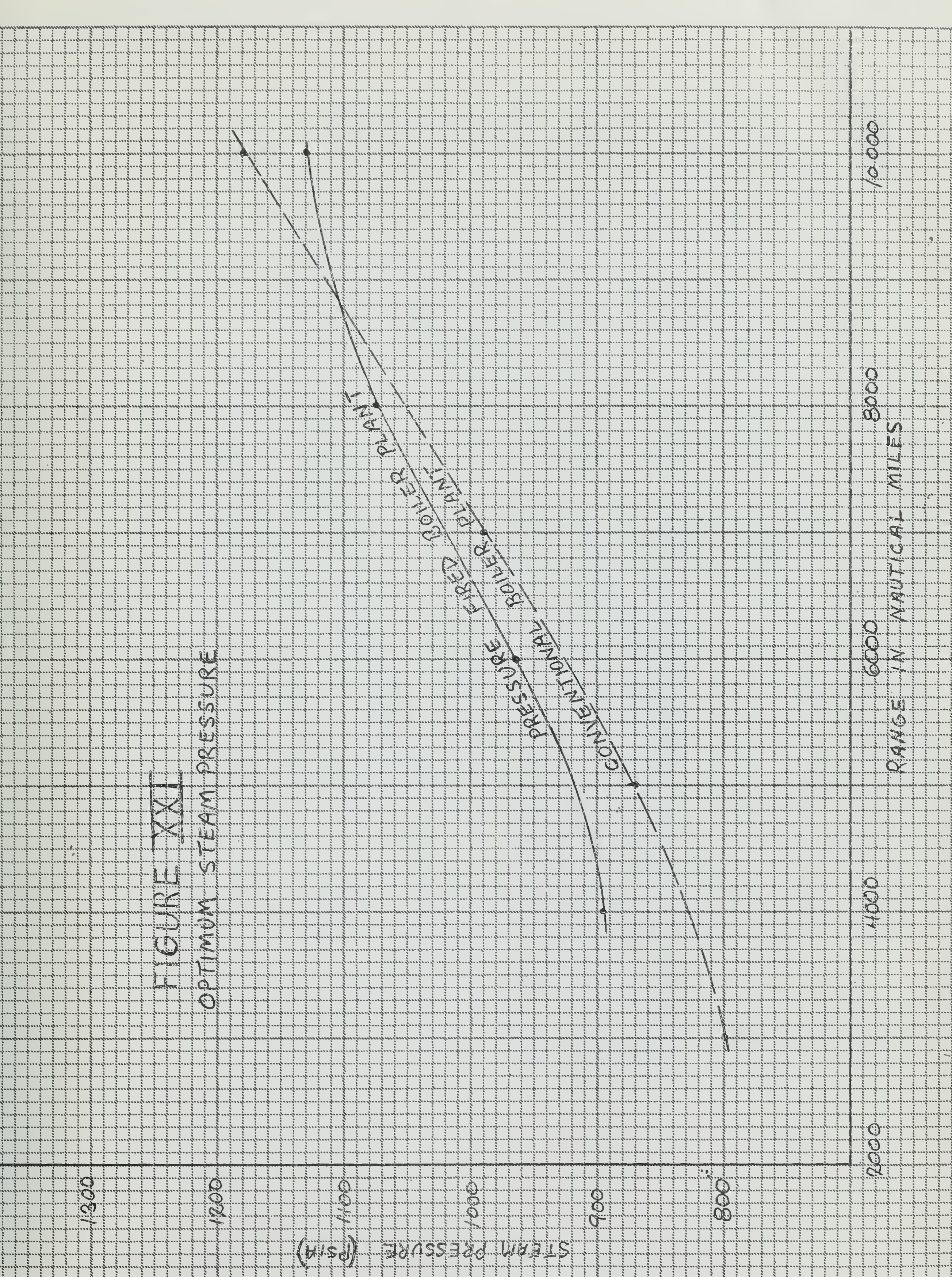


FIGURE XXI
OPTIMUM STEAM PRESSURE

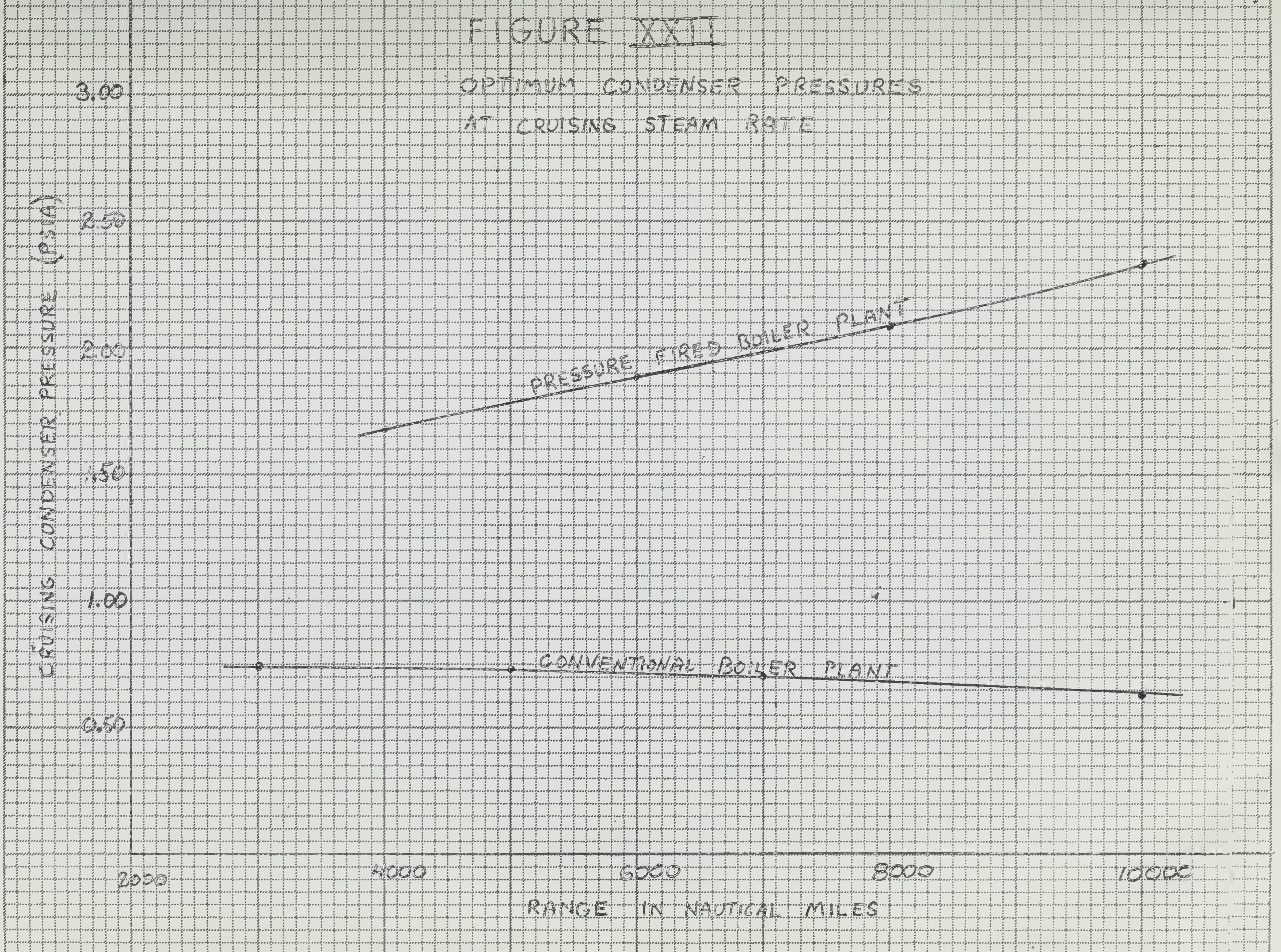


FIGURE XXII
OPTIMUM CONDENSER PRESSURES
AT CRUISING STEAM RATE
PRESSURE FIRED BOILER PLANT
CONVENTIONAL BOILER PLANT
CRUISING CONDENSER PRESSURE (PSIA)
3.00
2.50
2.00
1.50
1.00
0.50
2000
4000
6000
8000
10000
RANGE IN NAUTICAL MILES

FIGURE XXIII SCHEMATIC DIAGRAM OF PRESSURE FIRED BOILER

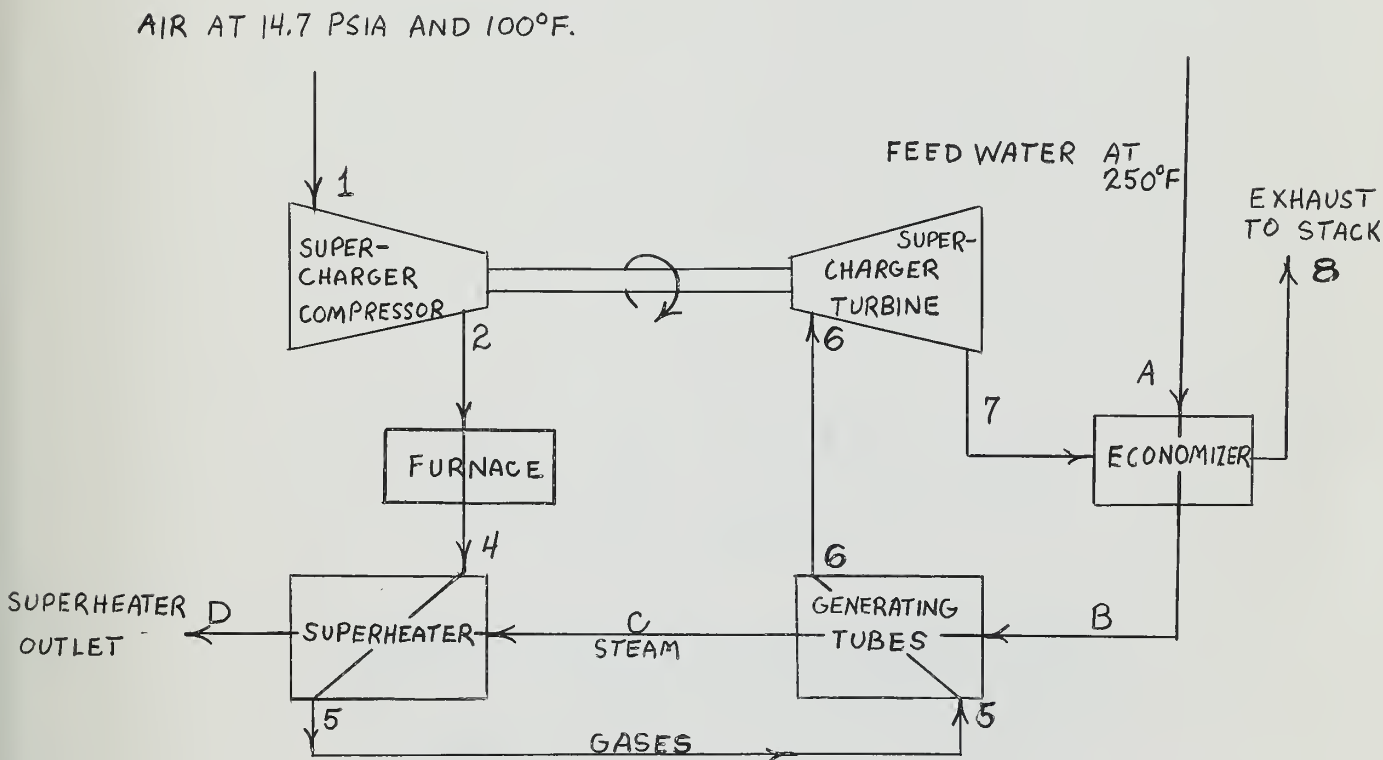


TABLE II BOILER WEIGHTS

P_s (LB/IN²)	PRESSURE FIRED BOILER WT. (LB)	CONVENTIONAL BOILER WT. (LB)	$\frac{(W_T)_{PF}}{(W_T)_{CONV.}}$
800	70,515	114,200	.616
1000	77,085	118,200	.651
1200	87,555	129,000	.679
1400	96,700	140,200	.689

TABLE III TOTAL OPTIMIZED PLANT WEIGHTS

RANGE IN NAUT. MILES	TOTAL WT. P.F.B. PLANT (LB.x10⁻³)	TOTAL WT. CONV. PLANT (LBx10⁻³)	WT. DIFF. (TONS)
4000	2137.0	2384	110.2
6000	2959.5	3260	134.0
8000	3779.0	4122	153.0
10 000	4592.0	4963	166.0

DISCUSSION OF RESULTS

The numerical results of this study show that the optimum pressure-fired boiler weighs about 60% of the corresponding forced draft boiler. This has been claimed by Daman and others, and is substantiated by table III.

The weights of the entire plant plus fuel is about 90% of that of a conventional plant, saving about 150 tons for a cruising range of 8000 miles.

The results show that the optimum furnace pressure and gas pressure drop are about 30 psia and 4.8 psi respectively, at the cruising rate. These correspond to a full power pressure of 100 psia, and pressure drop of 20 psi. These values are considerably higher than those designed into the Foster Wheeler boiler, and indicate that experimentation at these higher pressures and gas velocities is warranted. No geometrical restraints were imposed on the optimization problem. Increasing the furnace pressure and gas velocities allow a decrease in heat transfer surface. Just how this decrease is accomplished is a problem that has to be solved for each individual boiler design. It is shown in appendix E of this paper that practical designs are possible that take advantage of the reduction in heat transfer areas allowable by the optimum furnace pressure, without resulting in excessive tube metal temperatures and steam pressure drops in the superheater. It is therefore advantageous for the designer to experiment with boiler geometries that will be compatible with high furnace pressures and gas velocities.

The cycle incorporating the pressure fired boilers shows an increase in the optimum weight rate of steam flow over the conventional cycle. Along with this goes a marked increase in full power condenser pressure. This shift in the optimum condenser pressure can be reasoned out in the following manner. Increasing the condenser pressure has the general effect of increasing boiler weight, and decreasing the turbine and condenser weight. In conventional

cycles, the boiler weight is so predominant that very low condenser pressures are accepted in order to keep the boiler weight as low as possible. In the supercharged cycle, the drastic reduction in boiler weight places greater relative importance on turbine and condenser weight. Higher full power condenser pressures are necessary to trade off turbine and condenser weights for boiler weight.

Because of the wide range over which the parameters varied, and because all computation was done by hand, a minimum number of data points were used. To find exact values of minima and maxima, a more detailed study in a much narrower range of interest may be warranted. This study is not intended to be a boiler design, but serves the purpose of pointing out the need of further investigation and experimentation in the region of the optimum.

CONCLUSIONS

1. The design parameters of a pressure fired boiler, furnace pressure, gas pressure drop through the boiler, gas temperature at entrance to gas turbine, and steam pressure, all may be optimized to yield a minimum total weight of propulsion plant plus fuel for a given range, when such boiler is used in a modern destroyer steam plant.

2. The optimum furnace pressure is in the vicinity of 30 psia when operating at cruising steam rate in the DLG-6 steam plant. This corresponds to a full load pressure of about 100 psia.

3. The optimum gas pressure drop for the boiler in this plant is about 4.8 psi at the cruising rate, which corresponds to a full load drop of 20% of the furnace pressure, or 20 psi.

4. The optimum steam pressure varies from 895 psia for a range of 4000 nautical miles to 1130 psia for a range of 10,000 nautical miles. This is slightly higher than the optimum steam pressures for a plant with conventional marine boilers (2).

5. The optimum pressure fired boiler for this plant is about 40% lighter than the corresponding conventional marine boiler.

RECOMMENDATIONS

In the interest of developing compact, low specific weight marine propulsion plants, it is recommended that a much more detailed investigation of pressure fired boilers be undertaken, incorporating some actual designs. From the results of this study, it is recommended that any future investigations and experimentations be centered around the optimum values of the parameters developed in this study. The methods used are readily adaptable to digital computer programming, which would allow the use of many more data points than practical using hand computations.

APPENDIX

APPENDIX A

Details of Procedure

Furnace Temperature and Radiant Heat Absorption

Daman (5) gives the relationship

$$Q = KA \left[\frac{T_E}{1000}\right]^4 \tag{1}$$

where Q = radiant heat absorbed (BTU/hr)

A = projected radiant surface (ft^2)

T_E = furnace exit gas temperature (°R)

K = a constant dependent on heat input rate per unit of projected radiant surface.

From experiments with cylindrical water-cooled furnaces, a correlation of $K = 825$ for heat input rates of from 0.3 to 0.4×10^6 BTU/hr - ft^2; and $K = 1600$ for rates from 1.4 to 1.6×10^6 BTU/hr - ft^2 was calculated. (5) Gray (7) suggests the assumption of a constant furnace temperature in computing radiant heat transfer. Using this, $T_E = T_4$.

The radiant heat absorbed is equal to the heat input of the fuel, plus the sensible heat in the fuel before combustion, plus the sensible heat in the combustion air, minus the heat content of the gases at the furnace exit. Ignoring the heat in the fuel before combustion,

Q = (heat in air) + (heat of combustion) - (heat left in gases at furnace exit)

or

$$Q = q_1 + q_2 - q_3 \tag{2}$$

Looking at the supercharger compressor,

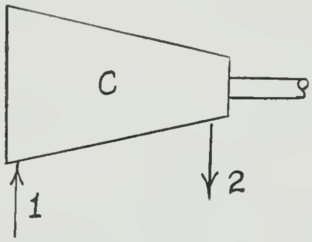
$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right) \exp. \frac{K-1}{\eta_{pc} K}$$

where η_{pc} is polytropic compressor efficiency, assumed to be .90.

$K = 1.4$

$T_1 = 560°R$

$p_1 = 14.7$ psia (atmospheric conditions)



$p_2 = 1.075\ p_4$ to allow for pressure drop between compressor discharge and furnace

$T_2 = 244\ p_4^{.3175}$

where p_4 is in psia

$C_p\ T_2 = 58.5\ p_4^{.3175}$ = heat in combustion air ($\frac{BTU}{lb}$)

q_1 = heat in combustion air ($^{BTU}/hr$) $= w_a\ C_p\ T_2$

w_a = air rate (lb/hr)

w_g = gas flow rate (lb/hr)

w_f = fuel rate (lb/hr)

$\frac{w_a}{w_f} = 15.8, \quad \frac{w_g}{w_f} = 16.8$ (see appendix D)

$$q_1 = 62.2\ w_g\ p_4^{.3175}\ (BTU/hr) \qquad (3)$$

Assume a higher heating value (HHV) of 19,100 BTU/lb. Lower heating value (LHV) is then 18,000 BTU/lb (see appendix D). Heat released in combustion, $q_2 = LHV\ w_f$. Assume boiler efficiency $\eta_B = .80 = \frac{G_b\ \Delta h}{HHV\ w_f}$

where G_b = steam rate (lb/hr)

Δh = gain in enthalpy through boiler.

So
$$w_f = \frac{G_b\ \Delta h}{.80\ HHV} \qquad (4)$$

The mean value of Δh in this range is 1300 BTU/lb, so (4) becomes

$w_f = .0855\ G_b$, and q_2 becomes $1540\ G_b$ (BTU/hr)

q_3 = heat left in gases at furnace exit = $w_g\ h_g$, or $w_g\ C_p\ T_4$

$w_g = 16.8\ w_f = 1.43\ G_b$

$q_3 = .457\ G_b\ T_4$.

Then (2) becomes

$$Q = 78.7\ G_b\ p_4^{.3175} + 1540\ G_b - .457\ G_b\ T_4 \qquad (5)$$

The value of KA in equation (1) may be estimated in the following way. Daman (5) found correlations for heat inputs of $.3$ to $.4 \times 10^6$ and 1.4 to 1.5×10^6 BTU/hr ft². The rated load of our model boiler yields a heat input rate of 1.5×10^6 BTU/hr ft². At cruising power this becomes approximately $.56 \times 10^6$ BTU/hr ft². Linearly interpolating, a good value of K at cruising rate in our model boiler is 965. Using the calculated radiant surface area of the model, KA = 93000. As furnace pressure increases, the volume required for complete combustion of the fuel decreases, and so radiant area decreases. An increase in pressure increases flame emissivity, with a resultant increase in the radiation constant K. It can be shown that the increase in K is of the same order of magnitude as the decrease in radiant area. The assumption was made that net effect is to keep KA nearly constant for all values of p_4. Equation (1) now becomes

$$78.7\ G_b\ p_4^{.3175} + 1540\ G_b - .457\ G_b\ T_4 = 93000 \left[\frac{T_4}{1000}\right]^4 \qquad (6)$$

Equation (6) is used to calculate furnace temperatures and radiant heat absorption rates for different values of p_4 and G_b.

Gas Temperature Drop Across Superheater

Writing a heat balance across the superheater yields

$$G_b\ (h_D - h_C) = w_g\ C_p\ (T_4 - T_5)$$

$$T_4 - T_5 = \frac{G_b\ (h_D - h_C)}{w_g\ C_p} = \frac{h_D - h_C}{.457} \qquad (7)$$

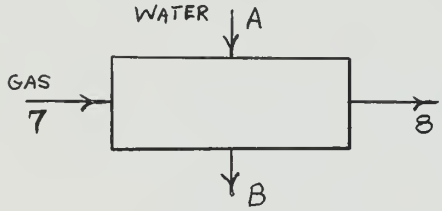
for cruising rate.

Economizer

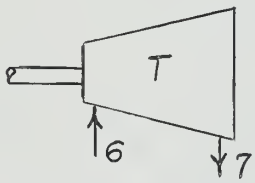
Assume a constant feed water temperature and enthalpy of 250°F and 222 BTU/lb.

$$w_g (h_7 - h_8) = G_b (h_B - h_A)$$

$$h_B - h_A = \frac{w_g (h_7 - h_8)}{G_b} = \frac{C_p w_g}{G_b} (T_7 - T_8) \tag{8}$$



To determine T_7, consider the supercharger turbine.



$$\frac{T_7}{T_6} = \left(\frac{p_7}{p_6}\right) \exp \eta_{pT} \left(\frac{K}{K-1}\right)$$

where η_{pT} = polytropic turbine efficiency, assumed to be .85

$K = 1.34$

$p_6 = p_4 - \Delta p$

$p_7 = 14.7$ psia

$$T_7 = T_6 \left(\frac{1.783}{(p_4 - \Delta p)}\right)^{.215} \tag{9}$$

Fix T_8 at 350°F at cruising steam rate to prevent condensation of acids in stack gases.

Then h_8 may be found by substituting (9) into equation (8).

$$h_B = h_A + \frac{C_p w_g}{G_b} \left[T_6 \frac{1.783}{(p_4 - \Delta p)^{.215}} - T_8 \right] \quad (10)$$

Generating Tubes

The heat balance across the generating tube bank is

$$G_b (h_C - h_B) - RH - DSH = w_g C_p (T_5 - T_6) \quad (11)$$

where RH = radiant heat transferred in furnace

DSH = heat transferred to boiler water during the desuperheating of auxiliary steam.

It is assumed that 7.65% of the output of the superheater section is desuperheated for auxiliary use. Therefore,

$$DSH = .0765\ G_b (h_D - h_C) \quad (12)$$

Substituting (12) into (11),

$$w_g = \frac{G_b \left[(h_C - h_B) - .0765 (h_D - h_C) \right] - RH}{C_p (T_5 - T_6)} \quad (13)$$

where T_6 is taken from table I.

Fuel Weight

$$W_F = \frac{w_g}{G_b} \times G_b \times \frac{w_f}{w_g} \times \frac{R}{V_K} \times 2 \quad (14)$$

where W_F = weight of fuel in pounds

R = cruising range in nautical miles

V_K = cruising speed in knots.

Relationships Between Cruising and Full Power Rates

For the destroyer plant under consideration, it was assumed the ratio of cruising steam rate to full power steam rate is .288.

$$\frac{(G_b)_{CR}}{(G_b)_{FP}} = .288$$

Full power is 120% rated capacity, cruising is 34.6% rated capacity.

From equations (4) and the value of 16.8 found for $\frac{w_g}{w_f}$ in appendix D, $\frac{w_g}{G_b}$ = 1.45 for cruising and 1.49 for full power. So, in going from cruising to full power, w_g increases by a factor of

$$\frac{(G_b)_{FP}}{(G_b)_{CR}} \times \frac{(G_b)_{CR}}{(w_g)_{CR}} \times \frac{(w_g)_{FP}}{(G_b)_{FP}} = 3.57$$

$$\frac{(w_g)_{FP}}{(w_g)_{CR}} = 3.57 \qquad (14a)$$

$$V \sim \frac{w_g\, T_g}{p} \qquad (14b)$$

where T_g is average gas temperature in boiler.
V is average gas velocity in boiler.

$\frac{(T_g)_{FP}}{(T_g)_{CR}}$ is about 1.04 in all cases

$$\frac{V_{FP}}{V_{CR}} = \frac{(w_g)_{FP}}{(w_g)_{CR}} \times \frac{(T_g)_{FP}}{(T_g)_{CR}} \times \frac{(p_4)_{CR}}{(p_4)_{FP}} \qquad (14c)$$

$\frac{(p_4)_{CR}}{(p_4)_{FP}}$ is obtained from the supercharger characteristics of the Foster Wheeler design.

$$\Delta p \sim p\, V^2$$

So, $$\frac{(\Delta p)_{FP}}{(\Delta p)_{CR}} \sim \frac{(p)_{FP}}{(p)_{CR}} \times \frac{(V)^2_{FP}}{(V)^2_{CR}} \qquad (14d)$$

The following relationships result:

$(p_4)_{FP}$	$(p_4)_{CR}$	$\dfrac{(\Delta p)_{FP}}{(\Delta p)_{CR}}$
40	19.4	6.71
60	23.1	5.30
80	26.8	4.62
100	30.5	4.19
120	34.3	3.94
140	37.9	3.70

Determination of T_6 required for self sustaining supercharger.

For self sustenance,

$$P_T = \frac{P_c}{\eta_m} \tag{14e}$$

where

P_T = turbine power

P_c = compressor power

η_m = mechanical efficiency

$$P_T = w_{gt}\, C_p\, T_6 \left[1 - \frac{T_7}{T_6}\right]$$

$$P_T = .28\, w_{gt}\, T_6 \left[1 - \frac{1.783}{(p_4 - \Delta p)^{.215}}\right] \tag{14f}$$

$$P_c = w_{gc}\, C_p\, T_1 \left[\frac{T_2}{T_1} - 1\right]$$

$$w_{gt} = 1.06\, w_{gc}$$

$$P_c = 127\, w_{gt} \left[.436\, p_4^{\ .3175} - 1\right] \tag{14g}$$

So, combining equations (14e), (14f), and (14g),

$$T_6 = \frac{130\left[.436\, p_4^{\ .3175} - 1\right]}{.28\left[1 - \dfrac{1.783}{(p_4 - \Delta p)^{.215}}\right]} \tag{14h}$$

APPENDIX B

Derivation of Weight Equations

Superheater

$$G_b (h_D - h_C) = U_T \, S \, \Delta T_m \qquad (15)$$

where U_T = total conductance (BTU/hr ft² °F)

S = heat transfer surface area (ft²)

ΔT_m = log mean temperature difference between steam and gas.

Assuming infinite conductance in tube metal,

$$U_T = \frac{(U_{rg} + U_{cg})\, U_{cs}}{U_{rg} + U_{cg} + U_{cs}} \qquad (16)$$

where U_{rg} = radiation conductance

U_{cg} = convection conductance, gas side

U_{cs} = convection conductance, steam side.

For the present, assume $U_{cs} \gg U_{cg}$, and assume that U_{rg} is a constant percentage of U_T.

Then $$U_T \sim U_{cg} \qquad (17)$$

Heat transfer correlation for turbulent flow on the outside of tubes gives

$$U_{cg} \sim \frac{k}{D} N_R^{.61} \qquad (18)$$

D = tube dia.

k = metal conductance

N_R = Reynolds number

$$U_{cg} \sim \frac{\rho^{.61} V^{.61}}{D^{.39}} \qquad (19)$$

where ϱ = density of gas

V = velocity of gas

From (8), draft loss $\Delta p \sim T_g \varrho V^2$,

where T_g is average gas temperature.

$p \sim \varrho T_g$

$\Delta p \sim p V^2$

$V^2 \sim \frac{\Delta p}{p}$

Substituting into equation (19),

$$U_{cg} \sim \frac{\varrho^{.61}}{D^{.39}} \left(\frac{\Delta p}{p_4}\right)^{.3} \sim \frac{p_4^{.3} \quad \Delta p^{.3}}{T_g^{.61} \; D^{.39}} \tag{20}$$

Re-arranging (15), $S = \frac{G_b (h_D - h_C)}{U_T \Delta T_m}$

Substitute for U_T, $$S \sim \frac{G_b (h_D - h_C) T_g^{.61} D^{.39}}{\Delta T_m \; p_4^{.3} \quad \Delta p^{.3}} \tag{21}$$

$W_{SH} \sim r \, l \, n \, t$

where W_{SH} = weight of superheater

r = tube radius

l = length of tube pass

n = number of tube passes

t = tube wall thickness

Geometrically, $S \sim r \, l \, n$.

So, $W_{SH} \sim S \, t \sim S \, p_s$ (22)

where p_s = steam pressure.

Substituting (18) into (19),

$$W_{SH} \sim \frac{p_s \, G_b (h_D - h_C) T_g^{.61} D^{.39}}{\Delta T_m \; p_4^{.3} \; \Delta p^{.3}} \tag{23}$$

Generating Tubes

$$U_T S \Delta T_m = w_g C_p (T_5 - T_6)$$

$$S = \frac{w_g C_p (T_5 - T_6)}{U_T \Delta T_m} \tag{24}$$

Following same reasoning as for superheater tubes, from equation (20),

$$U_T \sim U_{cg} \sim \frac{p_4^{.3} \Delta p^{.3}}{T_g^{.61} D^{.39}} \tag{25}$$

$$W_G \sim S p_s \tag{26}$$

Substituting (24) and (25) into (26),

$$W_G \sim \frac{p_s G_b (T_5 - T_6) T_g^{.61}}{\Delta T_m p_4^{.3} \Delta p^{.3}} \tag{27}$$

Headers and Downcomers

$$W_H \sim p_s r$$

where r is a linear dimension of an equivalent flow area.

$$G_b \sim r^2$$

$$W_H \sim p_s \sqrt{G_b} \tag{28}$$

Steam Drum and External Fittings

Assume simply $W_{SD} \sim p_s$ (29)

Furnace Pressure Shell

$$W_{PS} \sim p_4 l_F r_F + p_4 r_F^2$$

where r_F = radius of furnace, assume constant.

l_F = length of furnace, assume varies inversely as p_4.

$$W_{PS} \sim (K_1 + p_4) \tag{30}$$

where K_1 = a constant.

Internal Fittings

$$W_F \sim G_b \tag{31}$$

Supercharger and Associated Air Supply Ducting

The weight of rotary compressors and turbines can be considered proportional to the compressor or turbine work.

$$W_C \sim w_g \left(\frac{T_2}{T_1}\right) \sim w_g \left[\left(\frac{p_2}{p_1}\right)^{.3175} - 1\right]$$

$$W_C \sim G_b \left(.436\, p_4^{\,.3175} - 1\right) \tag{32}$$

Boiler Foundation

$$W_{FO} \sim W_B \tag{33}$$

where W_B = total weight of boiler.

Economizer

$$W_E \sim p_s S \tag{34}$$

$$S \sim \frac{w_g (h_7 - h_8)}{\Delta T_m U_T} \tag{35}$$

Assume total conductance U_T is constant for economizer. So (35) becomes

$$S \sim \frac{w_g (h_7 - h_8)}{\Delta T_m}$$

and (34) becomes

$$W_E \sim \frac{p_s w_g (h_7 - h_8)}{\Delta T_m} \tag{36}$$

Constant Weight

A constant weight of 2150 pounds is assumed, comprising the burners and burner wall refractory.

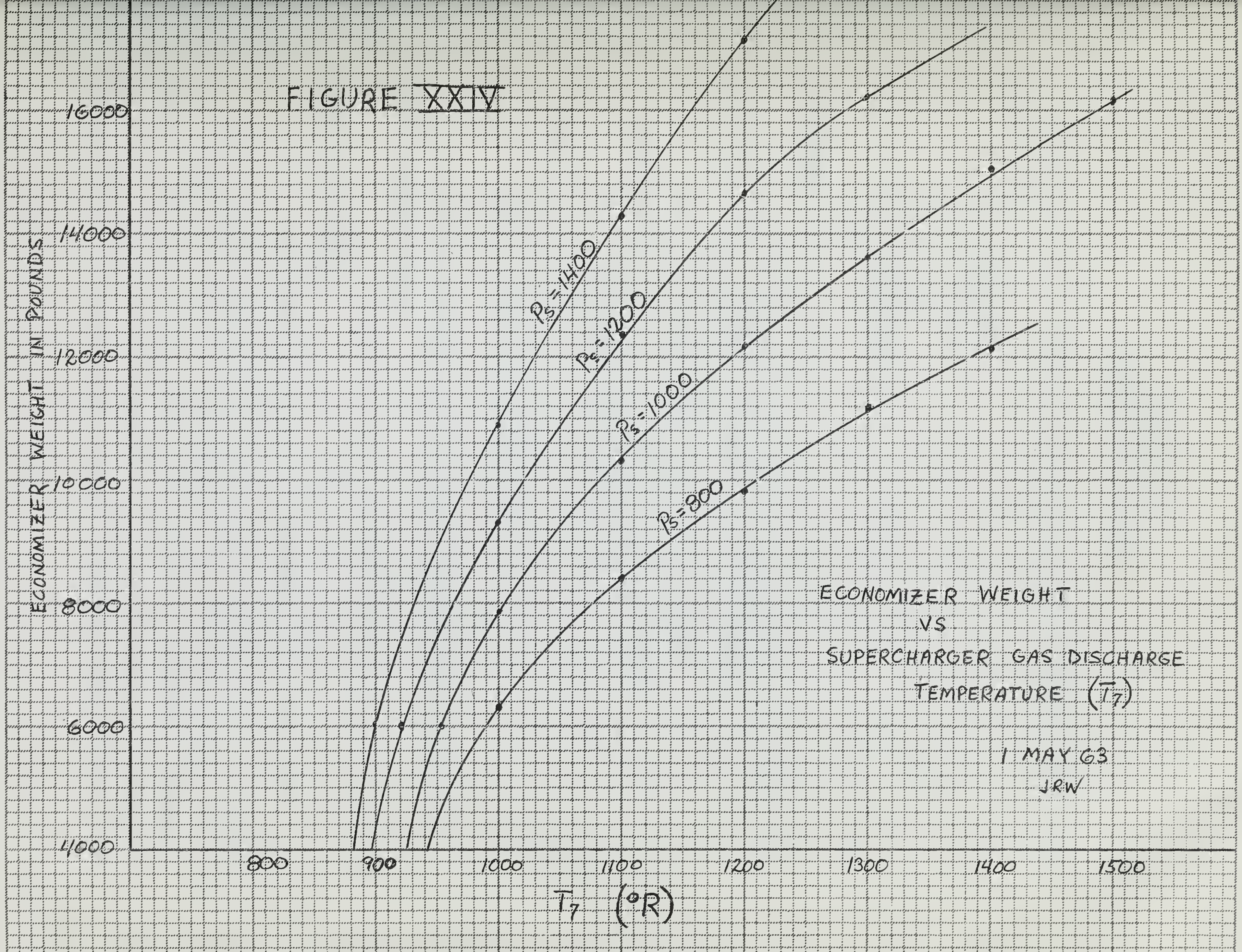


FIGURE XXIV
ECONOMIZER WEIGHT
VS
SUPERCHARGER GAS DISCHARGE
TEMPERATURE ($\overline{T}_7$)
1 MAY 63
JRW
$P_S = 1400$
$P_S = 1200$
$P_S = 1000$
$P_S = 800$
ECONOMIZER WEIGHT IN POUNDS
4000
6000
8000
10000
12000
14000
16000
$\overline{T}_7$ (°R)
800
900
1000
1100
1200
1300
1400
1500

APPENDIX C

Sample Calculations

Calculation of the weight of boiler plus fuel for the following case is done here in detail.

$p_s = 1000$ psia

$p_4 = 30.5$ psia

$\Delta p = 4.8$ psi

$G_b = 48{,}995$ lb/hr.

Fuel Weight

Substituting the values of p_4 and G_b into equation (6) and solving by trial and error gives $T_4 = 3362°R$, and $RH = 11.95 \times 10^6$ BTU/hr.

$h_D = 1533$ BTU/lb, and $h_C = 1192$ BTU/lb.

From equation (7), $T_4 - T_5 = 747$ °F, and $T_5 = 2615$ °R

$h_B = 284$ BTU/lb from equation (10).

Using $T_6 = 1105$ °R from table I,

$$\frac{w_g}{G_b} = 1.407 \text{ from equation (13).}$$

For a speed of 20 knots, $W_F = 410\ R$ from equation (14), where R is range in nautical miles.

Superheater Weight

Assume a tube diameter of one inch, the same as the model boiler.

Substituting the corresponding values from the model boiler into equation (23), a constant of 2.58×10^{-6} was obtained. Cruising rate pressures and temperatures and full power flow rates were used in all calculations. Cruising and full power flow rates are assumed to be linearly related; so it makes no difference which are used so long as one is consistent. Using 2.58×10^{-6} as

the proportionality constant in equation (23) yields a weight of 2660 pounds.

$$W_{SH} = 2,660$$

Generating Tubes

The proportionality constant was found to be 1.06×10^{-6}. Substituting into equation (27) yields

$$W_G = 11,190$$

Headers and Downcomers

Constant equals 2.3×10^{-3}. Equation (28) yields

$$W_H = 9510.$$

Steam Drum and External Fittings

Constant equals 12.5. Equation (29) yields

$$W_{SD} = 12,500.$$

Furnace Pressure Shell

Equation (30) becomes $W_{PS} = 3000 + 12.85\, p_4$.

$$W_{PS} = 4285.$$

Internal Fittings

Constant equals 8.8×10^{-3}. Equation (31) yields

$$W_F = 1500.$$

Supercharger and Air Ducting

Constant equals .115. Equation (32) yields

$$W_C = 24,900.$$

Economizer

Figure XXIV was plotted from equation (36). For the present case,

$$W_E = 6500.$$

Foundation

The weight of the boiler minus foundation adds up to 75, 205 lb. The constant of equation (33) is found to be .025, so

$$W_{FO} = 1880.$$

Adding all of the variable boiler weight plus the constant weight, and multiplying by the number of boilers, (75, 205 + 1880)4 = 308, 340

$$W_{4B} = 308,340 \qquad (37)$$

APPENDIX D

Combustion Calculations

Because of the higher furnace gas temperatures encountered in supercharged boilers, it is possible that an appreciable amount of ash will leave the furnace in liquid form. This would then plate out on the relatively cool generating tubes. For this reason, most of the experience with supercharged boilers to date has been with the use of distillate fuel.

The fuel assumed in this analysis was diesel oil of the following composition:

Carbon	85.5% by weight
Hydrogen	13.2%
Oxygen	1.1%
Sulphur	.2%

Higher heating value, 19100 BTU/lb.

$$.855C + .132H + .011O + .002S + X(.21)O + X(.79)N \rightarrow \text{products}$$

where X = lb air per lb fuel

$$.855C + .132H + (.011 + .21X)O + .002S + .79X\ N \rightarrow \text{products}$$

$$.855C + 2.28\ O_2 \rightarrow 3.135\ \text{lb}\ CO_2$$

$$.132H + 1.05\ O_2 \rightarrow 1.181\ \text{lb}\ H_2O$$

$$.002S + \underline{.002\ O_2} \rightarrow \underline{.004}\ \text{lb}\ SO_2$$

$$.989 + 3.332\ \text{lb}\ O_2 \rightarrow 4.320$$

$$3.332 = .011 + .21X$$

$$.21X = 3.321$$

$X = 15.8$ lb air/lb fuel is theoretical air for the fuel.

$4.32 + (.79)(15.8) = 16.8$ lb products per lb fuel for theoretical air.

With a pressured firebox, combustion of high efficiency is possible. The amount of excess air required should be a minimum. In the flow rate chart of reference (6), an air-fuel ratio of about 15.8 is indicated for the Foster Wheeler model. As shown above this represents zero excess air.

Following this, the assumption is made in this analysis of a constant air-fuel ratio of 15.8, resulting in 16.8 lb of products per lb of fuel burned.

$$\frac{w_g}{w_f} = 16.8 , \qquad \frac{w_a}{w_f} = 15.8$$

LHV = HHV - 1040 (lb H_2O formed per lb fuel burned)

LHV = 19100 - (1040)(1.18) = 19100 - 1220 = 17,880 BTU/lb.

APPENDIX E

Restrictions Placed on Design

There are many practical considerations that enter into any boiler design that place restrictions on how close the designer may approach the optimum operating parameters. Among these are limitations on maximum tube metal temperatures, maximum acceptable flow velocities with the resultant pressure drops, adequate gravity head for circulation (if natural circulation design), and other geometrical, heat transfer, and structural limitations. None of these have been considered yet in this study. They have been left out of the main body of the analysis not because they are unimportant in boiler design, but because they are functions of individual designs and do not enter into the general optimization of parameters.

The tube metal temperature is more likely to be critical in the superheater tubes because of the increased resistance to heat transfer that the film on the steam side shows over that of the boiling water in the generator tubes. The average rate of heat transfer through the superheater surface is $\frac{G_b(h_D - h_C)}{S}$.

It was previously assumed that 80% of this was by convective heat transfer, and 20% by direct and inter-tube radiation.

$$U_T \Delta T_m = .8 \frac{G_b (h_D - h_C)}{S} \quad \text{BTU/hr ft}^2$$

$$U_T = \frac{.8 G_b (h_D - h_C)}{\Delta T_m S} \qquad (38)$$

It is assumed that the total convective heat conductance U_T is constant along the superheater tubes. This only assumes uniformity in the tube walls and the fluid flow, which is a good assumption.

Heat transfer surface is

$$S = K \frac{W_{SH}}{p_s}$$

from equation (22),

where K is the proportionality constant to be determined

from the model boiler. Its value is found to be 141. So

$$S = \frac{141\ W_{SH}}{p_s} \tag{39}$$

The local heat flux density at any point is

$$Q_L = U_T\,(T_g - T_s)\ \text{BTU/hr ft}^2 \tag{40}$$

where T_g = local gas temperature

T_s = local steam temperature

and U_T is determined by substituting (39) into equation (38).

$$Q_L = \frac{Tw_L - Ts_L}{\frac{1}{U_w} + \frac{1}{U_{cs}}} \tag{41}$$

where Tw_L = max. wall temperature at a point L

Ts_L = steam temperature at a point L

U_w = conductance through wall

U_{cs} = steam film conductance

Assume that $U_w \gg U_{cs}$. Then from equation (41),

$$Tw_L = Ts_L + Q_L \frac{1}{U_{cs}} \tag{42}$$

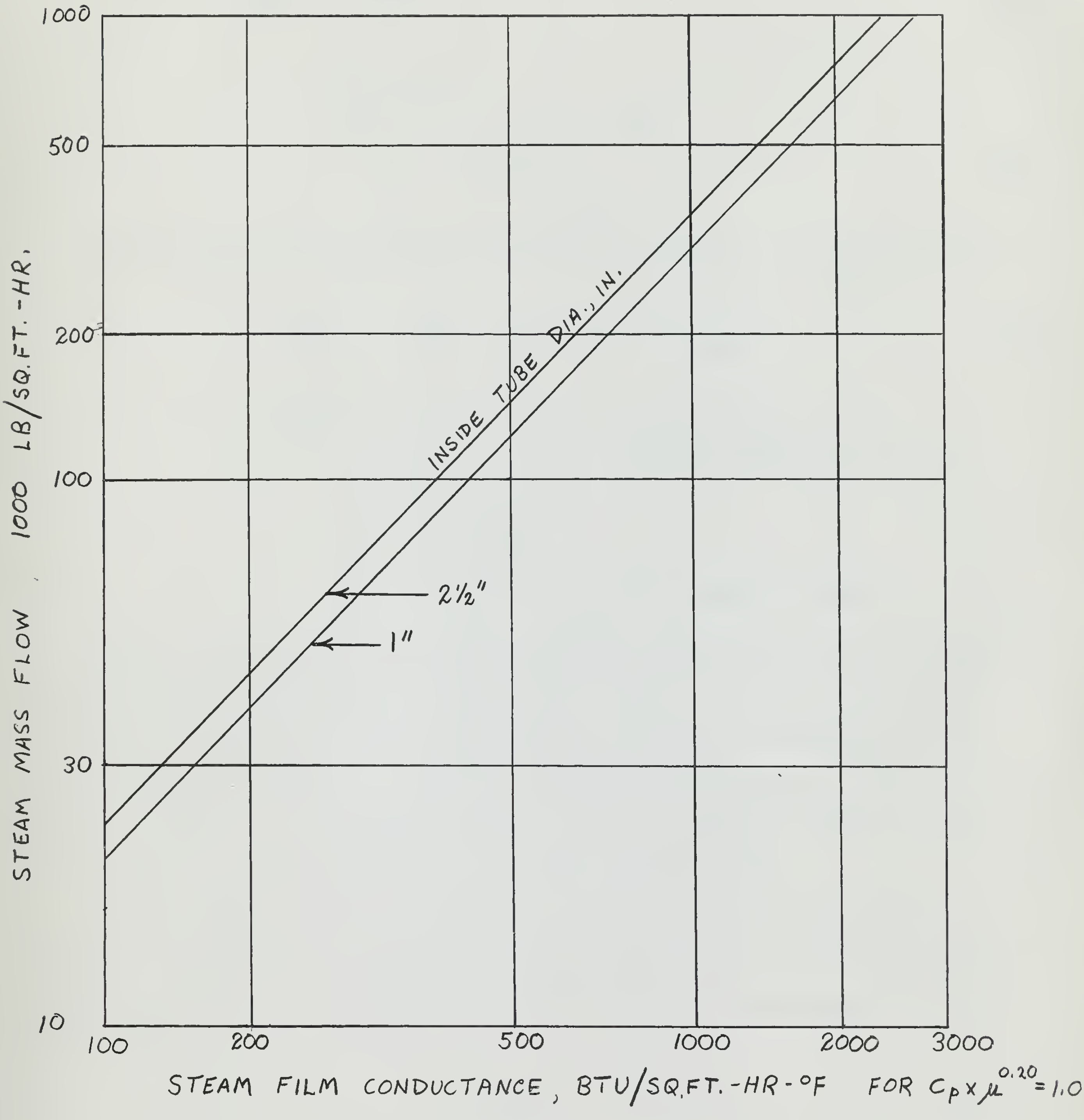
If an upper limit is set on Tw at any point, the required minimum value of U_{cs} can be determined from equation (42). Figure 3 on page 12-6 of reference (8) is a chart of G_b/A versus U_{cs}. Using this, the minimum steam flow rate may be determined that will keep Tw below the specified maximum. This chart of G_b/A versus U_{cs} is reproduced herein as figure XXV.

From a given boiler flow rate G_b, one can now determine the maximum allowable steam flow cross section (A_c) for the superheater.

$A_c = \pi r^2 n$

where r = tube radius

FIG. XXV THE EFFECT OF STEAM MASS FLOW ON STEAM FILM CONDUCTANCE. REPRODUCED FROM [8].



n = number of tube passes

$$n_{max} = \frac{A_{max}}{\pi r^2} \tag{43}$$

$$S = 2 \pi r n l$$

where l = tube pass length

$$l_{min} = \frac{S}{2 \pi r n_{max}} \tag{44}$$

Steam pressure drop through the superheater is given in reference (8) as

$$\left(\frac{1.5 v}{12} + \frac{f l v}{d} + \frac{n_b v}{12} \right) \left(\frac{G_b}{A} \quad \frac{1}{100000} \right)^2 \tag{45}$$

where v = average specific volume of steam

f = friction factor, a function of Reynolds number.

d = tube diameter

l = tube length

n_b = a factor to account for losses in bends.

Table IV lists some of these design considerations for the optimum steam rates and heat transfer surface areas found for each of four steam pressures. A maximum allowable tube metal temperature of 1150°F is postulated, and a superheater tube radius of 0.5 inches is assumed. The table shows that designs are feasible in the region of optimum furnace pressure without encountering excessive tube temperatures or steam pressure drops.

Boiler height and steam generating surface area are two parameters that determine whether natural circulation is feasible in a boiler designed for a certain steam rate. For balanced natural circulation, from page 9 - 5 of reference (8),

$$\frac{Z(\rho_1 - \rho_2)}{62.4} = F_d + F_r \tag{46}$$

where Z = vertical distance in feet from water level in steam drum to lowest point in circuit

TABLE IV

HEAT TRANSFER AND STEAM FLOW RELATIONSHIPS IN SUPERHEATER

	P_s (LB/IN^2)	G_b (LB/HR)	ΔT_m (°F)	U_T (Btu/HR-FT^2°F)	Q_L EXIT (Btu/HR-FT^2)	U_{CS} EXIT (Btu/HR-FT^2°F)	$(G_b/A)_{MIN}$
(1)	800	192 500	1665	84.6	121 500	1215	360 000
(2)	1000	188 000	1655	82.8	118 000	1180	347 500
(3)	1200	184 000	1632	81.0	115 000	1150	340 000
(4)	1400	182 000	1620	80.3	113 200	1132	330 000

N ≡ NO. OF TUBE PASSES ℓ ≡ LENGTH OF TUBE PASS

D ≡ DIAMETER OF CIRCULAR TUBE NEST δP_s ≡ STEAM PRESSURE DROP

	$(A_c)_{MAX}$ (FT^2)	$(N)_{MAX}$	$(\ell)_{MIN}$ (FT)	D_{MIN} (ASSUMING CIRCULAR TUBE NEST) FT.	V EXIT (FT./MIN)	V AVE. (FT/MIN)	δP_s (LB/IN.^{2})
(1)	.535	98	14.45	4.60	6520	4970	8.8
(2)	.542	99	14.50	4.62	4990	3780	6.7
(3)	.542	99	14.70	4.68	4040	3050	5.4
(4)	.554	101	14.80	4.72	3340	2490	4.4

ρ_1 = density of fluid in downcomers

ρ_2 = mean density of fluid in risers

F_d = sum of fluid flow losses in downcomers in feet

F_r = sum of fluid flow losses in risers in feet

$$F_d + F_r \sim \left(\frac{w}{A}\right)^2 \tag{47}$$

where w is flow rate in circuit

A_c is cross sectional flow area.

$w \sim G_b$, boiler steam rate

$A_c \sim \pi r^2 n$

$S \sim 2\pi r l n$

where l is length of generating tubes.

n is number of generating tubes.

$$A_c \sim \frac{Sr}{l}$$

So from (47), $F_d + F_r \sim \dfrac{G_b^{\,2}\, l^2}{S^2 r^2}$

From (46), $$Z(\rho_1 - \rho_2) \sim \frac{l^2}{S^2 r^2} \tag{48}$$

for a given steam rate.

Re-arranging equation (48),

$$S^2 \sim \frac{l^2}{r^2 Z(\rho_1 - \rho_2)} \tag{49}$$

From equation (49) it can be seen that for a given required steam rate, lowering the surface area of the generating tubes by increasing furnace pressure may require that the designer raise Z, the over-all height of the boiler, in proportion to the square of the reduction in surface area, if balanced natural circulation is to be maintained. Of course a given design may allow for changes in the other variables of equation (49), such as increasing tube diameter, decreasing the length (l) of the generating tubes, or decreasing the fluid flow losses in the circuit. In any case, retaining a natural circulation design will require some ingenuity from the designer.

APPENDIX F

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